COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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New power plant of Industrial Rayon Corporation at Painesville, Ohio; see page 24

Digest of Power Papers at A.S.M.E. Annual Meeting

Standards for Turbine-Generators

Power Plant of Industrial Rayon Corp.

A Method for Determination of Reactions and Stresses in Expansion Pipe Bends

COMBUSTION

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VOLUME TEN NUMBER SIX

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FOR DECEMBER 1938

FEATURE ARTICLES

EDITORIALS

DEPARTMENTS

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ALFRED D. BLAKE, *Editor*

THOMAS E. HANLEY *Circulation Manager*

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EDITORIAL

Standardized Steam Conditions and Capacities

At no previous period has there been such wide variation in conditions applying to the installation of power equipment as has obtained during the last five or six years. This may be attributed partly to progress, motivated by a desire to achieve optimum operating economy, and partly to the fact that the bulk of such equipment has gone into the modernization or extension of existing plants rather than new stations; hence it has been necessary to fit the new equipment to existing conditions and limitations. The result has been a preponderance of so-called "tailor-made" installations.

During a period of relative slackness in construction and moderate load demands, such practice is not serious from the standpoint of deliveries, but during a time of marked activity to meet urgent load demands, and particularly in the event of a national emergency, it might have grave consequences. It was with this in mind that the National Defense Power Committee recently initiated the preferred standards for turbine-generators, as listed elsewhere in this issue. Obviously, these standards, dealing with capacities and steam conditions, will be reflected in associated power equipment such as boilers, appurtenances and certain auxiliaries. However, because of the wide variation in fuel characteristics, it will be impracticable, beyond conforming to the specified capacities and steam conditions, to carry out standardization with steam generating units to the same extent as will be possible with turbine-generators and certain auxiliaries.

It will be much easier to apply the preferred standards to the design of new stations than to the modernization and extension of old plants, and to central stations than to industrial plants where process steam requirements may be found to complicate the attainment of a favorable heat balance. However, as the present standards do not apply to turbine-generators below ten thousand kilowatts, the number of industrial plants affected will not be great. Later, it is planned to extend the standards to units of smaller ratings. This would affect a greater number of installations.

That the Committee appreciates such contingencies is apparent from its statement that deviations which can be fully justified will be permissible in special cases. Moreover, the Committee has been careful to point out that the standards shall not be permitted to impede technical advances, to which end periodic reviews are suggested to determine whether advances in the art of power generation call for modifications in the standards.

A similar step was undertaken in Germany two or three years ago in which a committee of the Verein deutscher Ingenieure and the German Standards Commission, together with representatives of manufacturers and users, established standards for steam pressures and temperatures with the object of reducing initial costs.

The proposed standards in this country were drawn up by representatives of the utilities, the turbine manufacturers and the Government. While not mandatory, they have been accepted by agreement among the parties represented, which presages their acceptance by the power field within the limitations mentioned. They may entail certain compromises in plant design, occasionally perhaps at a slight sacrifice in estimated economy, but this is likely to be offset by some decrease in engineering expense and accelerated delivery of certain equipment, particularly the turbines.

High Pressures in the Marine Field

In marine circles there have long been two groups of opinion respecting steam pressures and temperatures. One favors emulating, to a certain degree, stationary practice in which higher pressures and temperatures have become well established. The other, more conservative, group feels that conditions surrounding land and marine operations are not sufficiently comparable and experience in the latter field too limited in this respect to warrant more than a moderate increase in pressure to around 400 or 450 lb, which is now being widely employed on the newer vessels. If credence be given to several recent items in the daily press, such a division of opinion exists in naval circles with reference to the new building program.

While several very high-pressure installations have been made in the foreign merchant marine, notably 1850 lb, 890 F on the Italian liner *Conte Rosso* and 1325 lb on the German freighter *Pottsdam*, these may be regarded as pioneer installations, despite reports of satisfactory performance. However, the moderately high pressures of 735 lb on the German liners *Scharnhost* and *Gneisenau*, and the freighter *Argo*, as well as 850 lb on the Dutch steamer *Kertosona* may be regarded as more indicative of trends abroad.

In the American merchant marine the new turbine-electric tanker *J. W. Van Dyke*, a vessel of 23,898 tons, employs a steam pressure of 600 lb and a temperature of 825 F. Its performance was discussed in a paper at the Annual Meeting of the A.S.M.E., which is abstracted briefly in the report of the meeting in this issue. From the figures given, it would appear that the economic performance is somewhat better than that attainable with diesel engines, considering the price differential between boiler and diesel oils.

Although this vessel has been in service for only a comparatively short time, there are no indications in the paper of any difficulties attributable to the higher pressure. Attention will be focused on its continued performance which undoubtedly will exert some influence on further trends in American marine practice.

Digest of POWER PAPERS at

AMONG a large number of technical papers presented at the Annual Meeting of The American Society of Mechanical Engineers, December 5 to 9, that held particular interest for steam power engineers, were those dealing with industrial power, cracks in a high-pressure boiler drum, pipe stress problems, modern boiler furnaces, the influence of steam-flow metering equipment on piping, high-temperature steam experience at Detroit, the utilization of pulverized coal ash, reheat factors, high pressure and temperature in the marine field and an investigation of fuel-bed conditions on a large multiple-retort stoker. Several other interesting papers pertaining to problems in the steam plant were presented but were not printed, hence were not available for abstracting.

Industrial Power

C. W. E. Clarke, consulting engineer with United Engineers & Constructors, Inc., presented an extensive review and discussion of power generation in the industrial field, together with references to certain utility installations that have bearing on industrial plant practice.[1]

Although maximum steam conditions up to about 1300 lb and 950 F can be justified in special cases, from the practical viewpoint, 900 lb and 850 F appear to be the sensible limits for most industrial plants. Exceptions noted were the Rouge Plant of the Ford Motor Company, employing the condensing cycle and 1215 lb, 925 F, and the Dow Chemical Company with superposition at 1250 lb, 825 F.

Steam cycles most commonly employed include high-pressure turbines exhausting to the existing low-pressure system, back-pressure turbines with one-point extraction to process, double-extraction condensing turbines, suitable for new plants, and mixed-pressure turbines. A number of prominent industrial plants employing such arrangements were described. Among these were the 10,000-kw, 800-lb, 800-F, superposed plant of the Weirton Steel Company; the 650-lb, 650-F installation of the Gulf Refining Company at Port Arthur, Texas; the 625-lb, 720-F units of the General Foods Corporation at Battle Creek; the 900-lb, 750-F plant of the Pittsburgh Plate Glass Company at Barberton, Ohio; the 800-lb, 750-F superposed units at the Akron plant of The Goodyear Tire & Rubber Company; the 900-lb, 750-F condensing plant of the RCA Manufacturing Company at Camden, N. J.; and the new plant of the Carnegie-Illinois Steel Corporation, Chicago, containing three 300,000-lb per hr, 450-lb, 750-F steam generating units supplying a 25,000-kw turbine-generator and three 75,000-cfm condensing turboblowers.

Steam pressure-reducing and desuperheating equipment, as developed for topping units in central stations, is finding new applications in industry. In case of outage of the high-pressure unit, continuity of operation of the low-pressure equipment is essential, hence availability and speed of response of the pressure-reducing and desuperheating equipment is important. In a large central station where the topping unit was out of service for several months, the pressure-reducing-desuperheating system was in almost continuous service for nearly 5000 hr. Under these conditions the heat rate was 6000 Btu per net kwhr higher than normal operation; yet the use of this emergency equipment resulted in a heat rate about 10 per cent lower than that which would have been obtained by using the old standby low-pressure boilers.

The trend in design of steam-generating units is toward larger capacities, higher steaming rates and more

[1] Because of the length of this paper a more comprehensive abstract will appear in a subsequent issue of COMBUSTION.

TABLE I—INDUSTRIAL-PLANT TURBINES INSTALLED IN 1936 AND 1937 LISTED ACCORDING TO TYPES

		Types of Turbines				
Throttle Pressure, Lb per Sq In.	Total Number of Turbines	Condensing	Condensing Extraction	Back Pressure	Back-Pressure Extraction	Mixed Pressure
1200	5	0	1	3	0	0
700–900	15	0	5	7	3	0
500–700	13	0	4	7	2	0
400–500	58	3	26	16	11	2
200–300	27	2	9	11	5	0
Total	117	5	45	44	21	2

TABLE II—TURBINES LISTED IN TABLE I GROUPED ACCORDING TO GENERATING CAPACITIES

	Capacity, Kw								
Throttle Pressure, Lb per Sq In.	Under 500	500 to 1000	1000 to 2000	2000 to 3000	3000 to 4000	4000 to 5000	5000 to 10,000	10,000 to 20,000	20,000 and up
1200	0	0	0	0	0	0	2	1	1
700–900	0	1	1	1	4	6	1	1	0
500–700	0	0	3	2	1	2	4	0	1
400–500	5	5	13	11	4	5	13	1	1
200–300	6	8	5	3	1	3	1	0	0
Total	11	14	22	17	10	16	21	3	3

TABLE III—OPERATING DATA ON HIGH-PRESSURE BOILERS AND TURBINES AT THE FORD MOTOR COMPANY UP TO MAY 1, 1938

	Boilers			Turbines		
Unit number	1	3	5	1 & 2	3 & 4	
Installed hours	60,264	56,568	15,672	59,332	15,648	18,275
In operation, hr	44,242	43,298	13,944	50,804	14,088	8,931
Down (no-load), hr	16,022	13,270	1,728	8,192	1,560	9,339
Outages, hr	0	0	0	336	0	3
Steam generated, 1000 lb	13,097,620	12,294,909	52,132,490	...	...	...0

A.S.M.E. ANNUAL MEETING

complicated automatic controls to permit utilization of several different fuels in the same furnace. In the latter connection was cited the furnaces at the Carnegie-Illinois Steel plant where tangential firing is employed for burning blast furnace gas as primary fuel and natural gas, oil and pulverized coal as secondary fuel; also a number of installations burning oil, gas or pulverized coal and plants in the paper-mill industry using wood waste and oil, gas or pulverized coal as supplementary fuel. Heat recovery from the burning of black liquor in kraft paper mills was also discussed at length.

The increasing use of self-contained boiler units for steam conditions up to 700 lb, 850 F and capacities up to 250,000 lb per hr was noted, and comments made on outdoor plants.

With reference to current practice as to industrial plant turbines, Table I lists units installed in 1936 and 1937, according to types, and Table II shows them grouped according to capacities. From this it will be noted that pressures of 400 to 500 lb are the most popular; there are only four machines listed for 1200-lb operation; most of the units are under 10,000-kw capacity; and mixed-pressure turbines are still rarely used.

Operating data were included on the high-pressure boilers and turbines at the Ford Motor Company's plant up to May 1 of this year. These are given in Table III.

Cracks in a High-Pressure Boiler Drum

Prof. A. E. White described the results of an investigation undertaken to determine the cause of cracks found in the forged drum of No. 5 boiler at the Edgar Station of the Boston Edison Company, also cracks in the water columns and in certain parts of the superheater fittings and headers of this boiler. The boiler is one of the first 1400-lb units installed in this country and had been in operation for approximately 60,000 hr at outputs ranging from 180,000 to 220,000 lb per hr since September 1927.

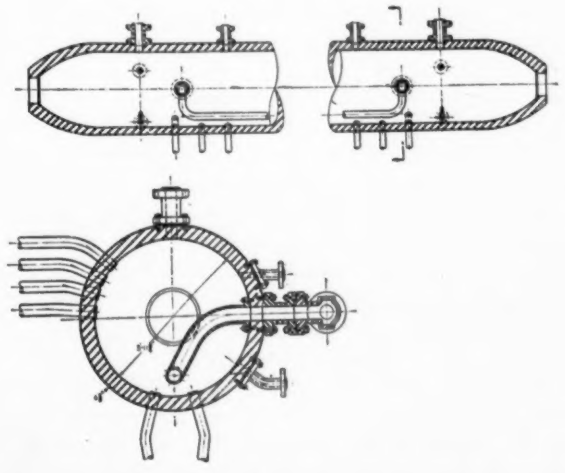


Fig. 1—Sections through drum showing original arrangement

Cracks of lesser magnitude had previously been discovered in the feedwater connections of No. 3 boiler but were not serious enough to warrant taking the unit out of service and subsequent checkups at six-month intervals had showed no extension of these cracks.

On No. 5 boiler, however, the cracks were in the walls of the drum forming part of the feedwater inlets, in the lower connections of the water columns, and in the forged-steel tee fittings at the inlet and outlet of the superheater, as well as in the flanges of the inlet and outlet superheater headers.

Sections through the boiler drum and the feedwater and water-column connections are shown in Fig. 1. It

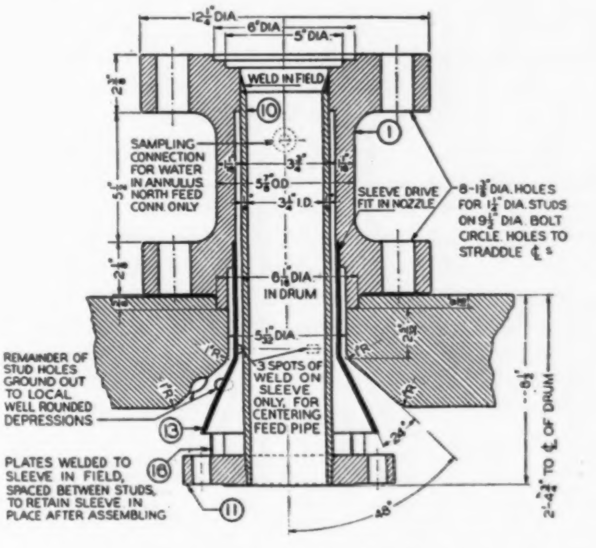


Fig. 2—Section of new inlet for feedwater

will be noted that the feedwater entered through a forged-steel nozzle attached to the drum by stud bolts and then passed through a hole in the drum to the internal feedwater distributing pipe which, in turn, was attached to the inside of the drum by stud bolts. This arrangement resulted in the bore in the drum becoming part of the channel through which the feedwater entered. The major cracks were found on the face of the bore.

Examination revealed that the cracks were transcrystalline, which fact precluded the possibility of caustic embrittlement. Moreover, their character indicated that they were not due to corrosion alone but had resulted from accelerated corrosion accompanied by repeated stress; that is, corrosion-fatigue.

The explanation of the repeated stress was found in the variation of the feedwater temperature which ranged from 250 to 350 F. Furthermore, at times when the feedwater regulator shut off the temperature of the water at the inlet would be that of the water in the drum, namely, 587 F. By having employed the bore of the drum as part of the feedwater channel the surrounding metal was subjected to repeated heating and cooling at an estimated rate of ten times an hour, or approxi-

mately 600,000 changes during the operating life of the boiler.

The solution to the problem involved changes so that the metal in the drum would not come in contact with the boiler feedwater during its passage into the drum, the new arrangement being as shown in Fig. 2. This necessitated boring out the cracked sections and attaching a tube to the outside flange of a new nozzle in such a manner that the walls of this tube would not come in contact with the wall of the drum. Also, to prevent any possible drip from the outside well of this tube onto the wall of the drum, an apron was attached to the inside. With this arrangement the temperature of the drum wall should be relatively constant and at the temperature of the water in the drum.

One of the possible contributing factors to the corrosion was the fact that the pH value of the water as it entered the drum varied from 6.9 to 7.2, while the water in the drum itself showed a pH varying from 11.0 to 11.7. The change in design resulted in freeing all sections of the boiler drum from contact with water having a low pH and substituting alkaline water with a pH of 11.0 to 11.7.

Water Columns

The corrosion-fatigue cracks found in the base of the water columns were believed to be due to the same causes as those which produced the cracks in the drum. To correct this condition, the inside edges of the male and female joints were rounded off for the purpose of removing stress concentrations. The most important change was the installation of a new connection 12 in. above the lower connection to provide a continuous circulation of water in the latter. With this arrangement the pH value in the lower section of the water column and in the connections to the water column would be raised from as low as 6.0 to values substantially the same as found in the drum, namely, 11.0 to 11.7. This change would also bring about a relatively constant temperature in the lower connections.

Superheater Fittings and Headers

It was believed that the small cracks found in some of the flanges of the forged-steel tee fittings on the inlet and outlet superheater headers, and in the headers themselves, were due to stress reversals in the presence of condensed steam. At these points pH values of probably less than 7.0 existed and would produce a corrosive condition. The remedy consisted of making such changes as would eliminate the collection of condensed steam and the removal of stresses by changing from a flanged to a welded construction. That is, forged-steel filler pipes were welded between the fittings and their respective headers after the flanges had been eliminated.

Pipe Stress Problems

H. W. Semar, mechanical engineer of the Westinghouse Electric and Manufacturing Company, in a paper entitled "The Determination of the Expansion Forces in Piping by Model Test" described apparatus used by that company for testing models of piping installations to determine the forces set up by expansion in the full-size installation. This is employed for checking the forces exerted on large steam turbines by the inlet and exhaust piping.

When the pipe lies in a single plane the calculation can be made readily, but when it is three dimensional the calculations become long and tedious. However, an accurate solution, with an error no greater than the experimental error in the measurement of the forces, can be made by tests of a scale model.

When a pipe expands with an increase in temperature, the forces exerted by its ends are the same as those produced by freeing one end, allowing it to expand, and then restoring the free end to its original position. The expansion forces can thus be duplicated by a displacement of one end of the pipe equal to the expansion but in the opposite direction. The problem is then one of load deflection and can be exactly described by a scale model.

The method consists of fixing one end of the model in a slide which can be moved a measured amount, and measuring the forces at the other end by means of a fixture which holds that end rigidly in place and permits measurement of six forces in coordinate planes, which are the equivalent of forces in coordinate directions plus moments in coordinate planes. The slide of the moveable end is placed so that its direction of motion coincides with the direction of the pipe expansion.

A complete example is given in the paper showing the steps necessary in determining the end reactions of an inlet pipe for a steam turbine.

This paper was followed by Progress Reports on "Creep Tests of Tubular Members" and on "Relaxation Tests."

Modern Boiler Furnaces

E. G. Bailey, vice president of Babcock & Wilcox Company, in a paper under the above title, discussed the effect of ash-fusion temperature and slagging on furnace design when burning pulverized coal, with particular reference to the two-stage furnace. Three boiler arrangements employing such a furnace were shown together with curves for each, representing gas- and ash-temperature measurements plotted against surface for the primary furnace, the secondary furnace and the boiler generating tubes. These measurements were taken with both high-velocity thermocouples and optical pyrometers the latter in all cases showing the lower values. Heat absorption in the primary furnace ranged from 35,500 to 83,600 Btu per sq ft per hr; in the secondary furnace from 21,000 to 30,500 Btu; and in the boiler tubes from 15,600 to 18,180 Btu. The temperature of gases leaving the primary furnace ranged from 2920 to 3100 F; those leaving the secondary furnace from 2500 to 2600; and those leaving the boiler tubes from 2120 to 2300. The heat liberation in the primary furnace ranged from 61,800 to 98,000 Btu per cu ft per hr, and that for the total furnace volume ranged from 31,800 Btu per cu ft per hr for a 600,000-lb per hr unit to 52,100 Btu for one of 300,000 lb per hr capacity.

The author observed that ash-fusion temperature determinations, as made in the laboratory, are not accurately indicative of the actual temperatures of the slag as formed in the furnace. Usually fusing temperature of the slag accumulated in the hotter zones near the burners and on the floor, or in the slag-tap section, is lower than that determined from the coal sample in the

laboratory and there is a general tendency for the slag and ash collected beyond the hot zone to be higher in all significant temperatures. Moreover, ash and slag having a wide spread between the initial deformation point and the fluid temperature are more likely to be troublesome than those with a smaller temperature range and a sharper melting point.

As to methods of measuring gas temperature, the high velocity thermocouple was stated to be the most accurate practical method, whereas a bare thermocouple is useful in measuring the temperature of slag particles floating in the gas stream. Certain forms of optical pyrometers, particularly those of the disappearing-filament type, have been found reasonably accurate for the highest temperature zones where uniform temperatures prevail.

Comparing Furnace Operation

The paper suggested a method of comparing furnace operating results by, (1) taking accurate gas temperatures throughout the furnace; (2) then calculating the rates of heat absorption in different zones; (3) calculating an "equivalent heat-receiving-surface temperature" for each zone; (4) taking samples of ash and slag from different portions of the furnace and determining the fusing-temperature range from such samples; (5) plotting these data as a "gas-ash temperature graph" against heat-absorbing surface; (6) comparing the data as shown on such a graph with observed behavior of the ash and slag in the furnace; and (7) comparing different furnaces, coals and methods of operation through the use of such graphs. Examples of the use of such a method were given.

The author supplemented his paper with colored motion pictures taken at different points in the furnaces and in the passes of several large steam generating units burning pulverized coal. These pictures showed the combustion and slag accumulations at the various locations, deslagging operations and disposal of slag from the continuous drip bottoms.

Influence of Steam-Flow Metering Equipment on Piping

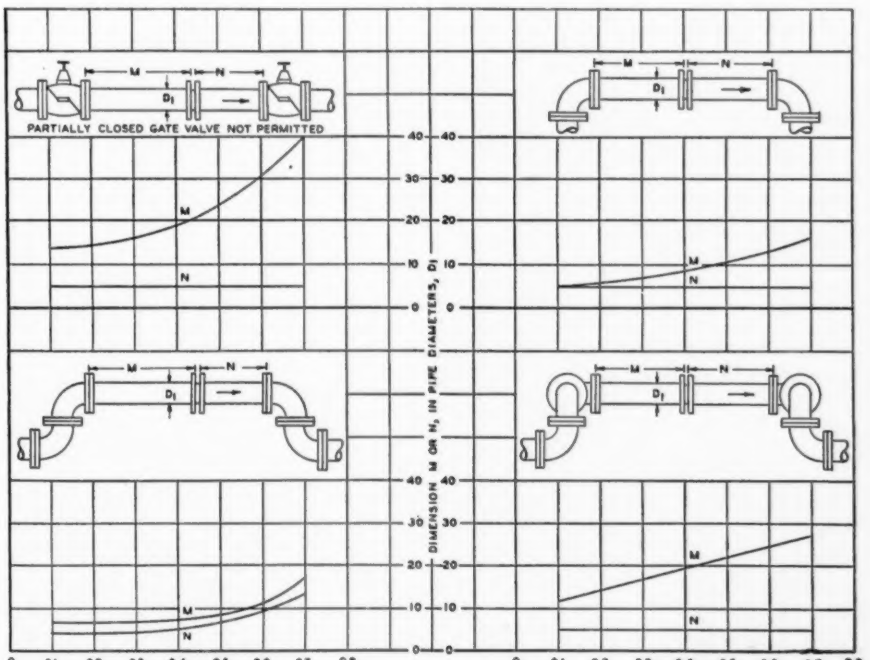
This paper by **R. M. Van Duzer, Jr.,** of The Detroit Edison Company, pointed out that the use of large capacity steam equipment in modern stations, with attendant increases in steam velocity, tends to complicate the problem of satisfactory metering and that the increasing use of flow measurements in place of water weights for acceptance tests emphasizes the necessity for a better understanding of the various factors influencing accuracy.

Two of these factors, directly related to piping design, are the ratio of the orifice-plate throat diameter to the inside pipe diameter and the location of the nozzle or orifice plate so as to permit satisfactory upstream and downstream flow.

If a diameter ratio much greater than 0.7 is used, flow through the throat is apt to be erratic, whereas too low a diameter ratio may result in a large net pressure drop and the use of an expensive high-differential-pressure secondary element. In view of this, the piping designer's problem then becomes one of selecting the economical line size based on the allowable overall pressure drop and the cost of the pipe, which should include the metering installation cost.

As to orifice plate location, while much is still unknown as to the disturbing influence of eddies and swirls resulting from pipe bends and fittings in the line ahead of the primary device, certain minimum conditions and values have been established by experiments. Some of these are represented in the accompanying figure.

The section of piping in which the primary element is installed should provide sufficient straight pipe upstream and downstream from the nozzle or orifice plate to eliminate swirls or eddies. Whenever there may be doubt as to whether undisturbed flow is obtained, the use of a straightening vane in addition to the minimum lengths of straight pipe is advisable, especially where the primary element is preceded by bends in more than one plane.



Minimum lengths of straight pipe for undisturbed flow without straighteners

For nozzle and orifice-plate installations, D_1 = inside pipe diameter and D_2 = diameter of throat of nozzle or orifice plate.

The condition of the pipe preceding the nozzle should also receive consideration. The inside surface of the pipe should be as smooth as possible and, if necessary, should be cleaned to remove mill scale or scale resulting from heat treatment. This is especially important with the smaller pipe sizes.

The author suggested the desirability of consulting with the meter manufacturers in the initial stages of piping design so as to avoid the necessity of supplying more expensive equipment to obtain a given range of accuracy. Also, it would be well for the meter manufacturers in installation instructions to list the magnitude of the probable metering errors involved, if all the various precautions are not taken in the installation of both the primary and the secondary elements.

The paper then described the piping and flow metering as employed in the rebuilt Conners Creek power plant and the extension to the Delray plant.

High-Temperature Steam Experience

It will be recalled that at the 1933 Annual Meeting, P. W. Thompson and R. M. Van Duzer, Jr., presented a paper on "High-Temperature Steam Experience at Detroit" in which was related the results of operation with 1000 to 1100 F steam in the 10,000-kw, 385-lb unit installed during 1930 at the Delray plant of The Detroit Edison Company. The present paper, under the same title, by Messrs. **Van Duzer** and **McCutchan,** relates the subsequent and final experience with this unit up to the time of its dismantling in 1937, after more than 26,000 hr of service, to permit rebuilding the turbine for superposed operation at 815 lb and 900 F, as part of the new extension to the Delray plant. The paper also deals with data obtained from the 1100-F experimental superheater and piping installation at the Trenton Channel plant.

The turbine operation has demonstrated the practicability of employing 1000-F steam. An examination of samples taken from all the principal materials used in the construction of the turbine, piping and superheater showed that, with a few exceptions, the alloys were all in good condition after this long period of operation. Measurements made to determine creep disclosed only small amounts and negligible deformations.

In the case of the 18-Cr, 8-Ni, 2-Si alloy steel tubing used in the superheater, which operated at 950 to 1150 F, the impact values were found to be low and the steel was susceptible to intergranular attack as a result of chromium-carbide formation, but the tubing still possessed good tensile properties. The 18-Cr, 8-Ni alloy castings, without the high silicon addition, were in excellent condition.

Few of the troubles encountered with the turbine were directly traceable to high-temperature operation except in the case of the turbine stop valve which had to be replaced, the bonnet flange being too light for the conditions. The replacement valve, designed to approximate the 900-lb standard, employed a ring-type gasket for the bonnet joint and gave reasonably good service, although it was necessary to remake the joint twice, at about 6000-hr intervals. The nitralloy bushings of the five control valves and the throttle valve, installed in place of the original special cast-iron parts, gave no trouble.

Trouble developed during the first two years of operation when putting the turbine on the line, due to vibration caused by shaft deflection. This condition, worse when the machine was still warm after a short shutdown, was partly corrected by more rapidly increasing the vacuum. Had the machine been equipped with a turning device, these starting troubles would not have been encountered.

During the latter part of 1933 the high-pressure shaft packing was found to be severely rubbed. This packing, of high- and low-tooth segmental design, was refitted with a total diametral clearance of 18 mils, after which the turbine gave no further trouble from vibration during the starting cycle.

Joints in the piping system that were not welded or provided with a welded seal gave some trouble and indicated the desirability of welded construction to insure freedom from periodic maintenance.

The experimental superheater and piping located in the Trenton Channel plant have operated for more than 48,000 hr, of which approximately 40,000 hr have been at 1100 F. Present efforts in connection with this equipment are being directed toward determining the steam-corrosion resistance of a group of steels and the creep properties of several accurately machined pipe sections.

The results of 1100-F steam-corrosion tests on eighteen steels, nine after 7500 hr of service and nine after 3800 hr, indicate that the scale formations offer material protection against subsequent steam attack and that the corrosion process is different in steam from that in air. Also, a comparison of creep determinations made on four machined pipe sections—two in service at 380 lb per sq in. and 1100 F, and two at 380 lb and 925 F—and laboratory specimens at the same temperature and stress, tend to disprove the theory that total diametral elongation of a pipe subject to internal pressure is materially less than that indicated by tensile-creep tests.

Utilization of Pulverized Coal Ash

The disposal of fly ash collected from pulverized-coal-fired boilers was discussed in a paper by **A. W. Thorson** and **John S. Nelles,** both of The Detroit Edison Company, their observations being based on investigations by that company over the past ten years in seeking a market for the fly ash collected at its Trenton Channel power house.

The most promising results from the standpoint of quantity disposal have been achieved in Cottrell block, in which the fly ash constitutes 90 per cent of the dry mix; in fly-ash-cinder concrete, where it completely replaces sand; in pebble concrete, as an admixture replacing up to 20 per cent portland cement; and in asphalt paving, as a filler.

For use in concrete, fly ash should contain not more than 13 per cent of the material retained on a 200-mesh sieve and not more than 7 per cent carbon.

Fly ash has also been used as a filler in paving materials, fertilizer, rubber, primer paint, puty, roofing material, common brick and cinder block. Comments on some of these uses follow:

Sheet Asphalt: In the laboratory it was found that a mixture of sand 84.5, fly ash 6 and asphalt cement 9.5 per cent by weight, would produce a paving material of satis-

factory workability, compressibility and stability. Not over 8 per cent ash should be used in any case.

Several sections of paving using fly ash have been laid in Detroit, the first, of about 1200 sq yd, having seen service for several years on a heavily traveled street without showing evidence of cracking, rutting or effects from freezing and thawing. Use of fly ash instead of portland cement as a surface filler for pavement is successful but because of the small quantity required is relatively unimportant unless the ash is also used as a filler in the mixture.

FERTILIZER: As a substitute for sand in fertilizers, fly ash appears attractive because of the quantity involved and the stability of the market, providing local conditions are favorable.

RUBBER: Fly ash as a filler in rubber, instead of fuller's earth at $5 to $12 per ton would involve considerable saving but preliminary tests indicated that it is too coarse and gritty for this purpose.

PAINT PRIMER AND PUTTY: Although a possible substitute for calcium carbonate, the color of the fly ash rendered it objectional for certain paint applications.

ROOFING MATERIAL: Shipments of fly ash have been made to some manufacturers of roofing products who report satisfactory results.

COMMON BRICK: Tests were made using 4 per cent fly ash in the mix for making 2000 common brick, but the results were very poor.

CINDER BLOCK: Fly ash has been used to form a dense waterproof outside face on cinder block.

Many uses of fly ash have been suggested as an aggregate in various products. Those investigated included sand-lime brick, aerocrete, light-weight concrete tile, acoustic plaster, Haydite concrete, lightweight concrete aggregate, sodium-silicate block, fly ash-cinder concrete and Cottrell block. Results in the first four mentioned were not satisfactory. Favorable results were attained with Haydite concrete but the market was small. While excellent results were had in using fly ash as an aggregate for light-weight concrete, the cost of the process rendered it prohibitive. An attempt was made to produce extruded building block from a mixture of fly ash and sodium silicate, but it set too rapidly, produced cracks and gave unsatisfactory compressive strength.

As a fine aggregate in cinder concrete the fly ash improves the workability and increases the strength. Also, because of its light weight such concrete is highly desirable for steel and concrete buildings. However, it has been prohibited by building codes because its porosity permits water to attack the steel.

The most promising use for fly ash has been in the manufacture of Cottrell blocks made from fly ash, lime and resin under patents controlled by Rostone, Inc. This block can be used in many types of construction both for exterior and for interior use, and it has been employed in the construction of a number of buildings around Detroit.

As an abrasive, fly ash has been tried as a substitute for pumice in metal polish, but the results were poor because it was not sufficiently abrasive. On the other hand, it has been used with success in the sandblasting of condenser tubes and turbine blades.

It has met with some success as a substitute for a portion of the clay in the manufacture of portland cement and in certain respects as an insulating material but has failed as a petroleum filter and as a foundry molding material.

Because of local conditions in the Detroit area, efforts have been concentrated toward development of projects using Cottrell blocks, fly ash-cinder concrete, concrete admixture and asphalt paving employing fly ash.

Thermodynamics Session

"Reheat Factors for Expansions of Superheated and Wet Steam" was the title of a paper by **Prof. C. G. Thatcher** of Swarthmore College. This related the results of an investigation to determine, from actual calculated condition curves, values of the reheat factor for expansions of superheated steam and wet steam, and to show a method of drawing the condition curve for expansions from the superheated into the wet region.

For superheated steam, charts were included showing the reheat factors plotted against the ratio $\frac{P_1}{P_2}$ up to a maximum value of 50. The effect of stage efficiency was considered. Wet region values were worked out by the step-by-step method and presented entirely separate from those in the superheat region. Here, also, the effect of stage efficiency were included, as well as the effect of decreasing stage efficiency due to increasing moisture content in the expanding steam. The lack of uniformity in practice on this latter issue was pointed out and a suggestion made which should lead to clearer understanding. Moisture removal, which is employed in many modern turbines, causes discontinuity in the condition curve, and a suggestion was incorporated for the construction of the curve for such a case. Finally, a method was given for calculating a condition curve for an expansion from any point in the superheat region and terminating in the wet region, but it was found impractical to present a chart of reheat factors for such expansions.

A second paper at this session on "Calculation of Steam-Turbine Reheat Factors" prepared by **R. B. Smith** of the Elliott Company, approached the reheat problem from the analytical standpoint. It derived relations for the reheat gain in the saturated- and superheated-steam regions, showing the influence of pressure ratio, temperature, efficiency and number of expansions. A comparison was made between the calculated reheat and the reheat computed graphically from the properties established by Keenan and Keyes, and good agreement was indicated over a wide range of conditions.

High Pressure and Temperature in Marine Field

The power plant of the turbine-electric tanker *J. W. van Dyke* was described and performance cited in a paper by **L. M. Goldsmith,** Chief Engineer of The Atlantic Refining Company. This 23,898-ton vessel is propelled at $13^1/_4$ knots by a 4500-kw, 3600-rpm turbine-generator taking steam of 600 lb gage, 825 F total temperature, and exhausting at 28.5 in. vacuum. The turbine is of the impulse type, with 15 stages, and bleeds after the third, seventh and twelfth stages. The pro-

pulsion motor is of the synchronous-induction type, 3-phase, 2300 volts, and develops 5000 shaft horsepower at 90 rpm.

Steam is furnished at 625 lb, 835 F, by two 22,500-lb per hr oil-fired boilers of the single-pass, straight-tube header type equipped with an air heater but no economizer. Each boiler has a water-heating surface of 3222 sq ft, a superheating surface of 1094 sq ft and an air heater surface of 2412 sq ft. With a furnace volume of 426 cu ft the maximum heat liberation figures 66,000 Btu per cu ft per hr.

Complete automatic combustion control is provided, including automatic superheat control. The superheater is divided into two sections and a desuperheating coil is located in the drum which is so baffled that the entering feed passes along this coil. The saturated steam from the drum passes through the first half of the superheater and then is divided by a three-way valve so that a variable portion goes through the desuperheating coil before passing on to the second half of superheater. The three-way valve is controlled automatically by the temperature at the superheater outlet. Charts reproduced in the paper show a very constant superheat under normal sea conditions.

The high-pressure piping employs stress-relieved welded joints, the main steam connections to the boilers, the boiler header and the line to the main turbine being of 0.5-Mo, 0.2-C seamless-steel pipe having an inside diameter of 5 in. and an outside diameter of 6 $^1/_2$ in. This is heavier than necessary but the Bureau of Marine Inspection did not permit taking full advantage of the greater strength of molybdenum tubing at high temperatures.

All the auxiliaries operate at boiler pressure. The circulating water for the condenser, instead of being taken direct from the sea, is taken from the double bottom which is vented. This reduces the velocity, provides deaeration and minimizes the chances of oxygen attack at the condenser tube entrances and also in the pump impellers. Also all heaters are of the vertical type with condenser-type heads to facilitate removal for cleaning. Flow meters and air-type soot blowers are installed.

Figures taken from a four-day voyage showed a fuel consumption for propulsion of 0.511 lb per shaft horsepower, neglecting fuel required by the auxiliaries. It is observed that when one considers the price differential between the fuel burned and that required for diesel engines, there should be no question as to the most efficient type of power plant for such a vessel.

Fuel Bed Tests at Hell Gate Station

An investigation of conditions in the full bed of a large multiple-retort underfeed stoker was reported in a paper by **M. A. Mayers, W. H. Dargan, Joseph Gershberg, B. C. Dalway, E. R. Kaiser** and **M. J. Williams.**

These studies were conducted jointly by the coal Research Laboratory of Carnegie Institute of Technology, the Consolidated Edison Company and Bituminous Coal Research, Inc., through its staff at Battelle Memorial Institute. The purpose was to determine temperatures, gas composition, the direction and magnitude of flow of both fuel and air in the fuel bed of a multiple-retort stoker. They were conducted on boiler No. 73 at Hell Gate Station of the Consolidated Edison Company, New York, which is a 165,000-lb per hr unit fired with a 14-retort, 37 tuyère stoker. Temperatures were measured by means of a probe and thermocouples.

The results were presented by charts representing cross-sections of the fuel bed on which were superposed contours of constant temperature, constant gas composition, and constant fuel-bed pressure. These were supplemented by motion pictures.

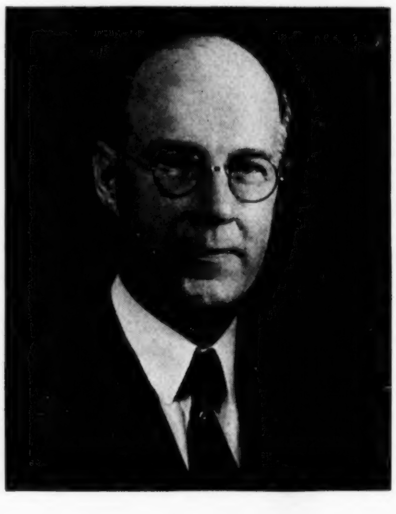
These charts showed that the flow of heat proceeds mainly in the horizontal direction, the coke layer forming first at the head end of the stoker, next to the burning lane, and progresses across the retort toward the center line. Carbonized fuel breaks off from the coke wall because of the formation of shrinkage cracks and the agitating action of the secondary rams, and falls down into the burning lane. These actions differ little with changes in load, excess air or character of coal.

Fuel bed temperatures and the gas composition depend largely upon the heating value of the coal, the heat capacity of the combustion air and the physical characteristics of the fuel bed.

A. G. Christie, New A.S.M.E. President

At the close of the Annual Meeting the new officers of the Society assumed their respective duties. The incoming president is A. G. Christie, Professor of Mechanical Engineering at The Johns Hopkins University, Baltimore.

Following graduation from the University of Toronto in 1901, Mr. Christie became associated with the turbine department of the Westinghouse Machine Company and left in 1904 to become an instructor and take graduate work for a year at Cornell University. From there he went with the Allis-Chalmers Company where he was engaged in the erection of many of its early turbines. In 1909 he joined the teaching staff of the University of Wisconsin and five years later was called to his present work at Johns Hopkins.



Aside from his educational duties, Professor Christie has acted as consultant for several private power companies and power commissions, notably the Hydroelectric Power Commission of Ontario and as chairman of the Commission on Power for the Province of Alberta, Canada. For some time he was advisory engineer with McClellan & Junkersfeld, Inc., in the design and construction of several large power plants, and on numerous occasions has been sent abroad to investigate and report on foreign power developments. His experience has been with both utility and industrial power.

Preferred Standards for Turbine-Generators

The National Defense Power Committee, appointed by the President several months ago to inquire into the adequacy of power supply as a measure of national defense, has released the report of its Subcommittee on Standardization which makes certain recommendations as to steam conditions and sizes of turbine-generators above 10,000 kw capacity. These are listed in the accompanying table.

From this table of preferred standards it will be noted that definite pressures, temperatures, vacuum and back pressure, voltage, power factor, etc., are recommended for nine sizes of condensing turbines, ranging from 10,000 to 100,000 kw and for eight sizes of superposed units from 10,000 to 60,000 kw. It is understood that the committee contemplates similar determinations for turbine-generators of smaller rated capacities to be announced at a future date.

The object in establishing these standards is to facilitate speed in the production of such power facilities to meet both peace-time and emergency conditions, through elimination of special designs and sizes. Incidentally, it is expected that the reduction in initial cost through such standardization will be sufficient to compensate for any sacrifice in economy by not employing special designs.

It is the intention that these standards will be followed in the construction of all new generating stations and in installations within existing stations unless the interrelation with existing equipment calls for special designs. With reference to the latter, the committee recommends that sufficient latitude be provided in the adoption of these standards to permit modifications to meet special conditions arising in connection with the installation of equipment in existing plants or where unusual physical conditions must be met. Provision is also made for the consideration of technical advances in the art. But in every case where deviation from the preferred standards is suggested, the burden of proof will be upon the manufacturer or user proposing such deviation.

Illustrative of cases where such deviation may become necessary are: (1) when the temperature of the condensing water supply may require turbines designed for back pressures higher than provided in the table; (2) if present availability in manufacturers' plants of turbine-generators well advanced in construction points to their completion and use; (3) duplication of a turbine-generator, not currently discontinued by the manufacturer as an active design, may expedite the manufacture and installation of power facilities; and (4) the manufacture and installation of turbine-generators adapted to meet essential requirements in existing stations may facilitate rapid creation of generating capacity.

These preferred standards for turbine-generators will be reflected in associated equipment such as boilers and appurtenances, steam piping valves, fittings and certain auxiliary equipment.

The committee making these recommendations was made up of representatives of the Government (including the Federal Power Commission, the Army and the Navy), the electric utilities and turbine manufacturers.

Preferred Standards for Steam Turbine-Generators

Prepared by Subcommittee on Standardization of the National Defense Power Committee

TABLE 1—CONDENSING TURBINES

GENERAL: All sizes—back pressure 1 in. or 1½ in. Hg abs.; short-circuit ratio 0.9; generator voltage 13,800; excitation voltage 250.

Rating, kw	10,000	12,500	15,000	20,000	25,000	35,000	50,000	75,000	100,000
Rpm	3,600	3,600	3,600	3,600	3,600	3,600	3,600	1,800	1,800
Throttle pressure, lb per sq in. gauge	650	650	650	850	850	850 or 1,250	850 or 1,250	850 or 1,250	850 or 1,250
Throttle temperature, F	825	825	825	900	900	900	900	900	900
Number of extraction openings	3	3	3	3	3	4	4	4	4
Temperature at extraction openings ±10 F (at rated output)	170/225/90	170/225/90	170/225/90	170/225/90	170/225/90	170/225/290 350	170/225/290 350	170/225/290 350	170/225/290 350
Turbine capacity in per cent of kilowatt rating	125	125	125	125	125	125	125	125	125
Power factor	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
Generator cooling (air or hydrogen)	Air	Air	Air	Air	Air or Hyd.	Hyd.	Hyd.	Hyd.	Hyd.

TABLE 2—SUPERPOSED TURBINES

GENERAL: All sizes—3600 rpm; Throttle pressure and temperature 1250 lb/sq in. gage, 925 F; Back pressure, 200–300 lb/sq in. gage. Short-circuit ratio 0.9; generator voltage 13,800; excitation voltage 250.

Rating, kw	10,000	12,500	15,000	20,000	25,000	35,000	50,000	60,000
Turbine capacity in per cent of kilowatt rating	111	111	111	111	111	111	111	111
Power factor	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8 or 0.9
Generator cooling (air or hydrogen)	Air	Air	Air	Air	Air or Hyd.	Hyd.	Hyd.	Hyd.

New Power Plant Serving Industrial Rayon Corporation

This new 14-area rayon plant at Painesville, Ohio, is served by a power plant containing 15,000 kw of turbine-generator capacity supplied with steam at 650 lb, 750 F, by three 90,000 lb per hr two-drum boilers fired with pulverized coal. A fly ash recovery system is installed. Hot water for process is obtained by extracting steam at 15 lb gage from the turbines. The manufacturing plant is completely air conditioned for which purpose 1500 tons of refrigeration is necessary and the water requirements for process and condensing amount to 15,000 gpm.

THE Industrial Rayon Corporation's new $11,500,000 plant at Painesville, Ohio, built for an annual production of twelve million pounds of custom-made rayon yarns by a new continuous spinning process, has lately gone into service and in certain aspects represents new achievements in industrial construction. Covering an area of fourteen acres, the manufacturing plant is a windowless structure containing 371,000 glass blocks and is completely air conditioned. For this purpose alone 1500 tons of mechanical refrigeration is required and the air conditioning system is divided into numerous zones, each thermostatically controlled to maintain the temperature and relative humidity within the narrow range suited to the process within the area served.

Substantial power facilities are necessary to serve such a manufacturing establishment with electricity, steam for process and heating, refrigeration, numerous pumps and other equipment. An exterior view of the power plant appears on the cover of this issue and interior views of the boiler room and pulverizer floor are shown herewith.

Power is supplied by three 5000-kw Westinghouse condensing turbine-generators taking steam at 650 lb throttle pressure and 750 F, with automatic extraction at 15 lb gage for process and heating. Condensing turbines of this size and extraction at 15 lb gage afforded the most advantageous heat balance which because of the continuous character of the process remains nearly constant except for seasonal variations. Any unbalance between power and process requirements, in the form of hot water, is taken care of by a large hot-water storage pond. Turbines driving certain of the auxiliaries also exhaust at 15 lb gage to the heaters.

Steam is furnished by three 90,000-lb per hr Combustion Engineering two-drum steam generating units designed for 725 lb steam pressure, 700 F total temperature at the superheater outlet. Each unit is equipped with a plate-type air heater, a bubble-type steam washer and fired with pulverized coal from two C-E Raymond bowl mills each of 6500 lb per hr capacity. The furnaces are completely water cooled having plain tubes on the front, sides and roof, as well as the bottom slag screen, and the burners, two per unit, are of the horizontal turbulent type.

Particulars of each of the steam generating units are:

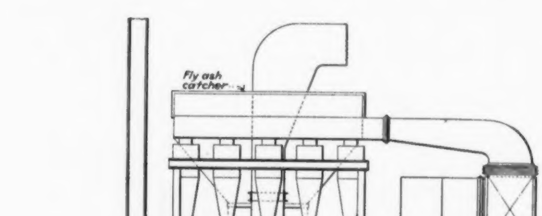
Water-wall surface	1820 sq ft
Boiler-heating surface	8700 sq ft
Superheater surface	2680 sq ft
Air-heater surface	10,560 sq ft
Furnace volume	4925 cu ft (gross)

There is one forced-draft and one induced-draft fan per boiler, two of each having dual turbine and motor drive and the third motor drive.

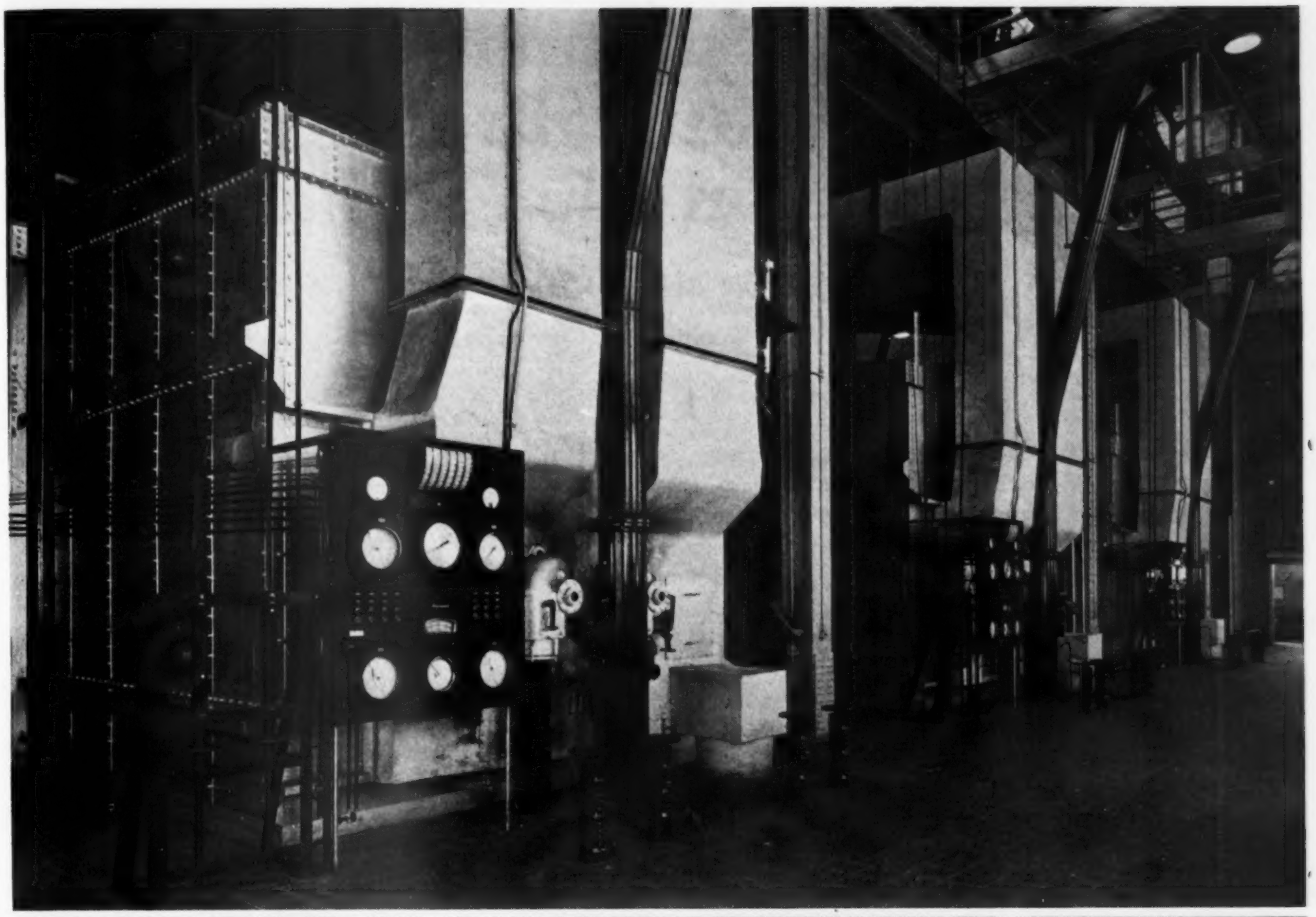
Fly ash recovery equipment of the Prat-Daniel multicyclone type has been installed over the boilers.

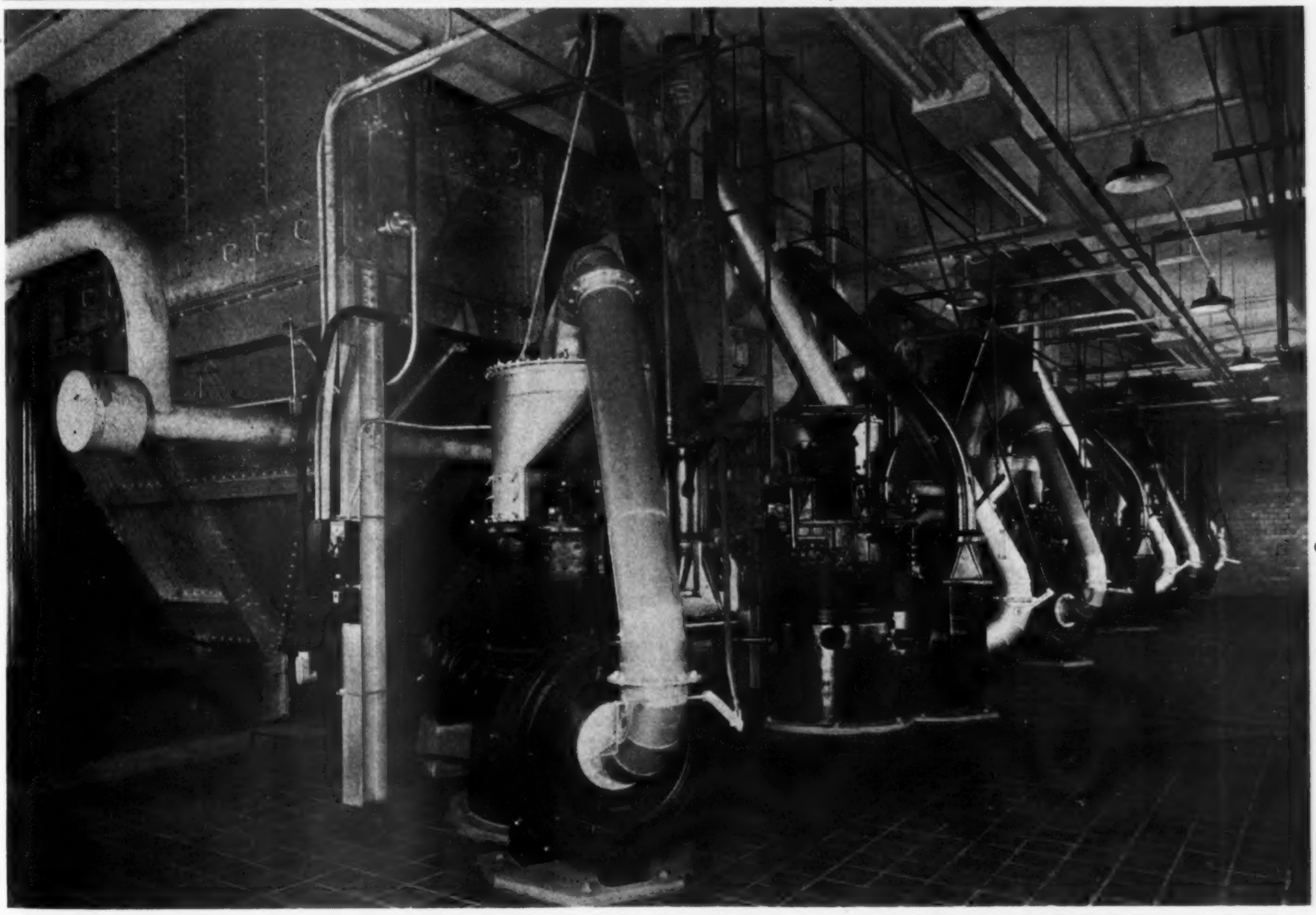
A vast quantity of water is required daily for process and for condensing. This is pumped from a large well 700 ft inland which, in turn, is fed by gravity through a 48-in. pipe line extending almost a mile into Lake Erie.

Part of the water from Lake Erie, which is handled by three 5000-gpm deep-well pumps, goes directly to the water treating system and part through the condensers serving the turbines and the ammonia compressors, thence to the treating plant. The hot water for process is controlled so as to maintain a temperature of 105 F.



Section through boiler unit





Upper view shows boiler operating floor and lower view the pulverizers

A Method for Determination of Reactions and Stresses in Expansion Pipe Bends

By F. PEITER AND DR. M. J. FISH

Combustion Engineering Company, Inc.

The use of higher temperatures and pressures in the design of steam generating equipment has caused considerable interest and investigation in the determination of the stresses and thrusts of piping subjected to a temperature change. In the design of the modern high-pressure boiler, many tubes are constrained at the ends and they must be provided with sufficient flexibility if the stresses are not to be excessive. Methods of solution of this problem, which depend upon exact mathematical analysis, are highly complicated, involve considerable work and are subject to numerical errors. The authors have evolved the following grapho-analytic method for the solution of the single plane pipe line. This method can be applied with ease to pipe lines and tubes which are unsymmetrical and where the shape is such that a numerical calculation would be hopelessly cumbersome.

SOME years ago Dr. A. Bantlin,[1] of Stuttgart, made tests on short-radius double-offset pipe bends similar to that of Fig. 1. These experiments were made on seamless steel tubing and on cast-iron bends. The results with seamless steel tubing showed deflections as much as five times that which could be accounted for by the ordinary formula for bending of curved solid bars. The tests on the cast-iron bends did not show such discrepancies. Bantlin ascribed the discrepancy between the observed and the calculated displacements in the steel tubes to wrinkles and corrugations in the pipe wall during the process of bending.

Shortly thereafter, Dr. Th. v. Kármán[2] developed formulas by the method of minimum energy which explained the discrepancy by the flattening of the circular pipe section which takes an oval form when loaded. This flattening is due to bending while the pipe is subjected to expansion forces and not to the process of manufacture. Kármán's equation was found independently by Hovgaard[3] who verified it by tests made on full-size specimens at the Massachusetts Institute of Technology in 1926. These two investigators accounted for the flattening effect at the arcs by the introduction of a "flexibility factor."

In the method described in this paper, tangent lines are substituted for the arcs. This gives sufficient accuracy in view of the various uncertainties present in the problem. Since the introduction of the flexibility factor at the arcs lowers the factor of safety of the pipe line and since the influence of the arcs is small, if the ratio of the straight lengths of pipe to the length of arcs is large, the factor has been neglected in the following work. The pipe line will be considered to lie wholly in one plane with its ends fully restrained and subjected to a uniform temperature change. As such, it constitutes a statically indeterminate system. When a pipe line is restrained from moving, the stresses and end reactions are increased. Therefore, the assumption of full restraint is on the safe side.

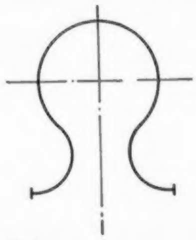


FIG. 1

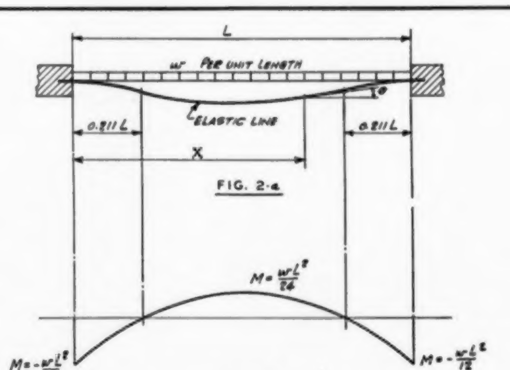


FIG. 2-a

FIG. 2-b

To review some elementary notions before analyzing the actual problem, consider a beam, restrained at the

[1] "Formänderung und Beanspruchung federnder Ausgleichröhren," A. Bantlin, *Zeit. V. D. I.*, vol. 54, no. 2, 1910, pp. 43–49.
[2] "Ueber die Formänderung dünnwandiger Rohre, insbesondere federnder Ausgleichröhen," Th. von Kármán, *Zeit. V. D. I.*, vol. 55, 1911, p. 1889.
[3] "The Elastic Deformation of Pipe-Bends," Wm. Hovgaard, *Jour. of Math. and Physics*, M. I. T., vol. VI, no. 2, 1926.

ends, and uniformly loaded, as in Fig. 2 (*a*). The elastic line or the curve of the neutral surface after bending will have the two points of inflection at a distance of 0.211L from the ends. The bending moment diagram of the beam Fig. 2 (*b*) shows that the value of the bending moment at the points of inflection is zero. It should be noticed that due to the restraint at the ends, the total change in the angle of flexure (the angle θ made by a tangent to the elastic curve with the horizontal) over the length of the beam is zero.

The fundamental equation of the beam theory is

$$\frac{d\theta}{dl} = \frac{M}{EI} \tag{1}$$

where, dl = an elementary length of the neutral surface of the beam.
$d\theta$ = the change in slope of the tangent to the neutral surface in the length dl.
M = the bending moment of the beam, which is usually a function of x only.
E = the modulus of elasticity of the material.
I = the moment of inertia of the cross-section of the beam.

Integrating equation (1),

$$\theta_L - \theta_0 = \int_0^L \frac{M}{EI}\, dl \tag{2}$$

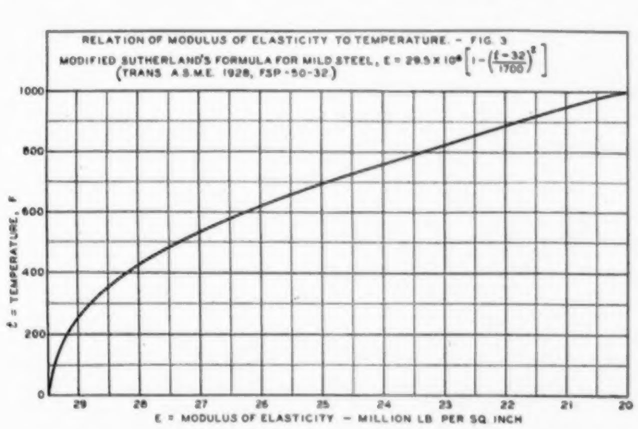
where, θ_L = the angle of flexure at the right end of the beam.
θ_0 = the angle of flexure at the left end of the beam.

The value of I will be taken as constant but E varies with the temperature according to the formula which is given for mild steel in Fig. 3. However, once the temperature t in degrees F is given, E is constant and will be taken so in the analysis. Equation (2) can be written as

$$\theta_L - \theta_0 = \frac{1}{EI}\int_0^L M\, dl. \tag{3}$$

It is seen that the expression $\int_0^L Mdl$ is the area of the moment diagram and it follows that the difference in slope between two points on a beam of uniform section is the area of the moment diagram between these points divided by EI. Therefore, it follows that the moment area is a direct measure of the angle of flexure along the beam.

In the analysis of the pipe line, the following assumptions are made: The cross-section of the tubes is circular and uniform; local stresses due to fabrication have been relieved; the dead weight of the members is neglected;



RELATION OF MODULUS OF ELASTICITY TO TEMPERATURE. — FIG. 3

the influence of the longitudinal forces (direct tension or compression) is neglected; the energy due to direct shear is negligible; the bending moment at any point of the members is not affected by the deflection of the members; and the material obeys Hooke's law.

Now, consider the symmetrical pipe line in Fig. 4, consisting of the two equal vertical legs, L_2, and the horizontal leg, L_1. The arcs have been replaced by ideal corners in order to simplify the demonstration. Let the ends A and B of the system move along parallel restrained paths under the action of the forces P. This

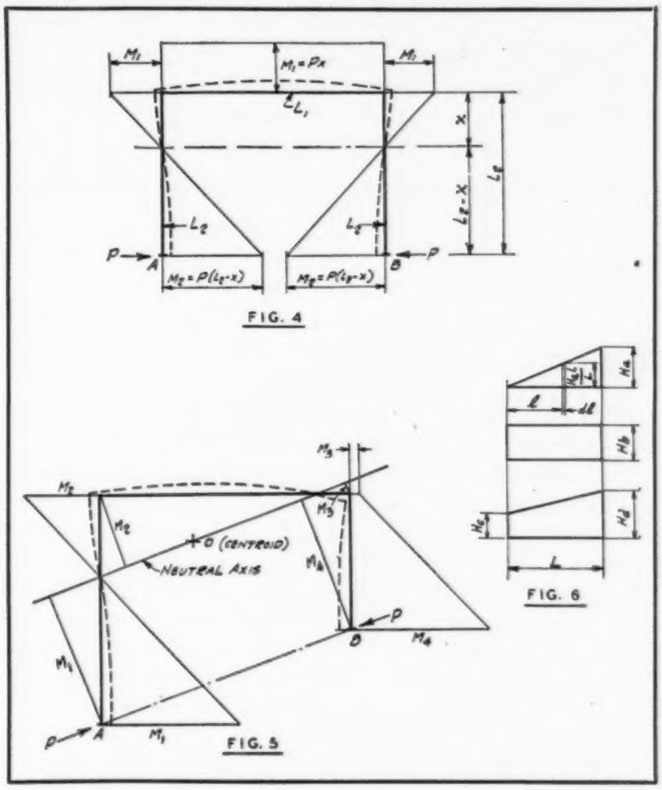


FIG. 4

FIG. 5

FIG. 6

means that the ends A and B will undergo the motion of translation but are restrained from rotation. The members of the system will flex as shown by the dotted line in Fig. 4. Since there is no rotation of the ends, the sum of all the angles of flexure (positive and negative) must equal zero. Therefore, it follows from equation (3) that the sum of the moment areas is also zero. This latter condition is true only when the moment diagram areas are taken about an axis parallel to the direction of translation of the restrained ends and through the centroid of the system. The moment at any arbitrary point of a member is equal to the force P times the lever arm h, where the latter is the perpendicular distance from the above mentioned axis through the centroid to the arbitrary point. Therefore, it follows that the moment is zero where the gravity axis intercepts the members of the system and since it is well known that where the moment is zero there is a point of inflection of the elastic line, then the points of inflection are determined by the intersection of the members with the gravity axis.

The flanges in the pipe line should be placed as near as possible to the points of inflection in order to prevent excessive bending on the bolts with consequent leaking at the gaskets.

To prove and illustrate the foregoing statements, draw in Fig. 4 a line parallel to AB and at any distance x from member L_1. Taking moments about this line, the moment in $L_1 = Px$ and is denoted by M_1; the moment in $x = Px$ and is denoted by M_1; and the moment in $L_2 - x = P(L_2 - x)$ and is denoted by M_2.

Since the total change in the angle of flexure in the members equals the moment area divided by EI, then

$$\Delta\theta_{L_1} = \frac{PxL_1}{EI}$$

$$2\cdot\Delta\theta_x = 2\cdot\frac{Px^2}{2EI} = \frac{Px^2}{EI}$$

$$2\cdot\Delta\theta_{(L_2-x)} = -2\left[\frac{P(L_2-x)^2}{2EI}\right] = -\frac{P(L_2-x)^2}{EI}$$

Where $\Delta\theta$, with the subscript, is the change in the angle of flexure of the member denoted by the subscript letter, and

$$\Delta\theta_{L_1} + 2\cdot\Delta\theta_x - 2\cdot\Delta\theta_{(L_2-x)} = 0 \qquad (4)$$

Substituting the values above, equation (4) now reads

$$\frac{PxL_1}{EI} + \frac{Px^2}{EI} - \frac{P(L_2-x)^2}{EI} = 0 \qquad (5)$$

Expanding (5)

$$\frac{P}{EI}\left[xL_1 + x^2 - L_2^2 + 2L_2x - x^2\right] = 0$$

Therefore,

$$x = \frac{L_2^2}{L_1 + 2L_2} \qquad (6)$$

The right-hand side of equation (6) is the expression of the center of gravity of the pipe line. This proves the statement that for zero moment area, the moment area diagrams must be taken about an axis parallel to the direction of translation and through the center of gravity. If the arcs are replaced by tangents instead of ideal corners, the effect is to change the position of the center of gravity of the system and the numerical values of the bending moments but the method of procedure will be the same. The accompanying examples will illustrate the use of the tangents to replace the arcs.

In a similar manner, it can be shown that the method above is applicable to asymmetrical systems. Fig. 5 shows such a system with the neutral axis parallel to the direction of translation of the ends A and B and passing through the centroid, O. The dotted line is the elastic line with its points of inflection or points of zero moment

THERMAL EXPANSION OF STEEL PIPE (FROM 70°F) IN INCHES PER 100 FT.

	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°
0°								0.00	0.067	0.154
100°	0.240	0.318	0.397	0.474	0.551	0.631	0.710	0.790	0.870	0.951
200°	0.033	1.113	1.194	1.278	1.362	1.444	1.525	1.608	1.692	1.777
300°	1.861	1.947	2.032	2.118	2.204	2.288	2.371	2.462	2.553	2.635
400°	2.717	2.813	2.908	2.995	3.082	3.176	3.270	3.356	3.442	3.540
500°	3.638	3.734	3.829	3.921	4.012	4.107	4.202	4.298	4.393	4.491
600°	4.589	4.684	4.779	4.874	4.969	5.071	5.173	5.268	5.362	5.466
700°	5.570	5.664	5.767	5.872	5.977	6.076	6.175	6.282	6.388	6.490
800°	6.592	6.694	6.806	6.905	7.004	7.117	7.230	7.335	7.440	7.548
900°	7.655	7.771	7.887	7.992	8.097	8.207	8.317	8.428	8.538	8.651

$$L_t = L_o\left[1 + a\frac{(t-32)}{1000} + b\left(\frac{t-32}{1000}\right)^2\right]$$

L_t = Length @ Temp t° F. (in feet)
L_o = " " 32° F. " "
$a = .006212$ $b = .001623$
FORMULA FROM SMITHSONIAN PHYSICAL TABLES. 7TH REV. ED. PAGE 218

FIG. 7
FOR MILD STEEL

again at the intersections of the neutral axis and the members of the system.

Having determined the points of inflection of the system, the internal potential energy of bending is to be equated to the external work producing the bending. From the theory of elasticity, it is well known that the internal work of bending is,

$$W = \frac{1}{2EI}\int_0^L M^2\,dl \qquad (7)$$

Equation (7) may be written,

$$W = \frac{P^2}{2EI}\int^L h^2\,dl \qquad (8)$$

where h is the variable lever arm of the bending moment and dl is an elementary length of the member. Referring to Fig. 6, in the integration of equation (8),

$$\left.\begin{aligned} W &= \frac{P^2}{2EI}\int_0^L\left(\frac{H_a l}{L}\right)^2 dl = \frac{P^2H_a^2L}{6EI}, \text{ for a triangle.} \\ W &= \frac{P^2H_b^2L}{2EI}, \text{ for a rectangle.} \\ W &= \frac{P^2L}{6EI}\left(H_c^2 + H_cH_d + H_d^2\right), \text{ for a trapezoid.} \end{aligned}\right\} \qquad (9)$$

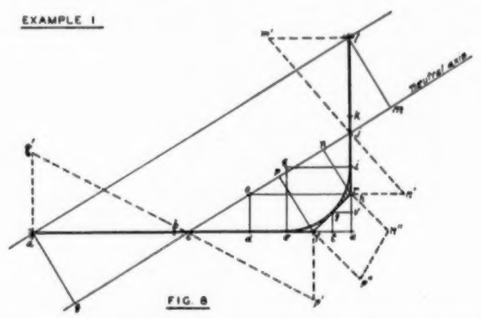


FIG. 6

In Fig. 4, denote the value of x as found by equation (6) by H_1 and $L_2 - x$ by H_2. Then from equations (8) and (9),

$$W = \frac{P^2}{2EI}\left[H_1^2L_1 + \frac{2H_1^2H_1}{3} + \frac{2H_2^2H_2}{3}\right] = \frac{P\Delta}{2} \qquad (10)$$

where P is the external force and Δ, the change in distance between A and B due to force P. Therefore,

$$P = \frac{\Delta EI}{\left[H_1^2L_1 + \frac{2H_1^3}{3} + \frac{2H_2^3}{3}\right]} \qquad (11)$$

If the method of least work or minimum energy is applied to equation 10, then from

$$W = \frac{P^2}{2EI}\left[L_1^2x^2 + \frac{2}{3}x^3 + \frac{2}{3}(L_2 - x)^3\right]$$

it follows that

$$\frac{dW}{dx} = \frac{P^2}{2EI}\left[2L_1^2x + 2x^2 - 2(L_2 - x)^2\right] = 0$$

and,

$$x = \frac{L_2^2}{L_1 + 2L_2}.$$

which checks the value in equation (6).

Illustrative Examples

In order to simplify the calculations in the following illustrative examples, the pipe in each case will be given an outside diameter of $3^1/_4$ in., a thickness of 0.240 in. and subjected to a temperature of 540 F. The moment of inertia, I = 3.58 in.[4], the temperature change, 540–70

= 470 F and the coefficient of thermal expansion equals 4.012 in per 100 ft. (Fig. 7).

EXAMPLE 1. Consider the pipe *aegil* of Fig. 8, fixed at the ends *a* and *l*, with the dimensions given. Draw the tangent *fh* and replace the original pipe by *aefghl*. Locate the centers of *af*, *fh*, *hl* at *b*, *g* and *k*, respectively. Then, find the center of gravity of the pipe line by taking moments about *af* and then about *hl*, as follows, slide rule accuracy only appearing in these computations:

Moments about *af*	Moments about *hl*
$af \times 0 = 106 \times 0 = 0$	$hl \times 0 = 58 \times 0 = 0$
$fh \times 7 = 20 \times 7 = 140$	$fh \times 7 = 20 \times 7 = 140$
$hl \times 43 = 58 \times 43 = 2494$	$af \times 67 = 106 \times 67 = 7102$
$184 \quad 2634$	$184 \quad 7242$
$\frac{2634}{184} = 14.3$ in.	$\frac{7242}{184} = 39.3$ in.

This locates point *o*, the centroid, such that od = 14.3 in. and or = 39.3 in.

Next, draw the neutral axis *qm* through *o* parallel to *al* cutting the members at the points of inflection *c* and *j*. The perpendicular distances *aq*, *fp*, *hn* and *lm* are the lever arms for the moments at points *a*, *f*, *h* and *l*, respectively. The moment area diagram consists of the four triangles $aq'c$, $fp'c$, $hn'j$, $lm'j$ and the trapezoid $fhn''p''$ with $aq = qq'$, $fp = fp' = fp''$, $hn = hn' = hn''$ and $lm = lm'$. Therefore, the value of the denominator of the right side of equation (11) can be found as follows:

$$\text{triangle } aq'c, \quad \frac{(30)^2 \times 58}{3} = \frac{900 \times 58}{3} = 17{,}400$$

$$\text{triangle } fp'c, \quad \frac{(25)^2 \times 48}{3} = \frac{625 \times 48}{3} = 10{,}000$$

$$\text{triangle } hn'j, \quad \frac{(20)^2 \times 23}{3} = \frac{400 \times 23}{3} = 3{,}067$$

$$\text{triangle } lm'j, \quad \frac{(30)^2 \times 35}{3} = \frac{900 \times 35}{3} = 10{,}500$$

$$\text{trapezoid } fhn''p'', \quad \frac{[(25)^2 + 25 \times 20 + (20)^2]}{3} \times 20 = \frac{1525 \times 20}{3} = \underline{10{,}167}$$

$$51{,}134$$

The value of E at 540 F is found as 26,900,000 from Fig. 3; Δ, the change in distance between *a* and *l* due to thermal expansion, has the following value,

$$\frac{140}{12} \times \frac{4.012}{100} = 0.469 \text{ in.}$$

The external force, P, can be found from equation (11).

$$P = \frac{0.469 \times 26{,}900{,}000 \times 3.58}{51{,}134} = 885 \text{ lb.}$$

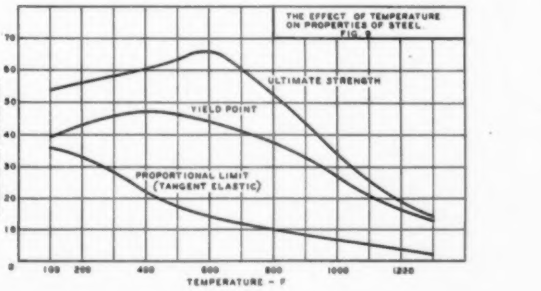
The maximum bending moment occurs at *a* and *l*.

$$M_a = M_l = 885 \times 30 = 26{,}550 \text{ in. lb.}$$

The fiber stress, S, due to the above moment is

$$S = \frac{26550 \times \frac{3.25}{2}}{3.58} = 12{,}050 \text{ lb per sq in.}$$

This latter value must be combined with stresses due to internal fluid pressure to obtain safe working values. To show how this is done, assume that the pipe lines in each of the examples are carrying steam from a boiler to a steam turbine, that the gage pressure is 365 lb per sq in.



The longitudinal stress in the pipe due to the internal fluid pressure can be found from the Lamé formula with Clavarino's correction, namely $S_1 = \frac{r_1^2 p}{3(r_2^2 - r_1^2)}$, where r_2 is the external radius, r_1 is the internal radius, p is the internal pressure and Poisson's ratio has the value $\frac{1}{3}$. Therefore,

$$S_1 = \frac{(1.385)^2 \times 365}{3[(1.625)^2 - (1.385)^2]} = 320 \text{ lb per sq in.}$$

This latter value will be the same in all the accompanying examples. The total longitudinal fiber stress then is

$$S + S_1 = 12{,}050 + 320 = 12{,}370 \text{ lb per sq in.}$$

The stresses due to the weight of the piping between hangers and supports, if the latter are properly designed, are usually small and may be neglected.

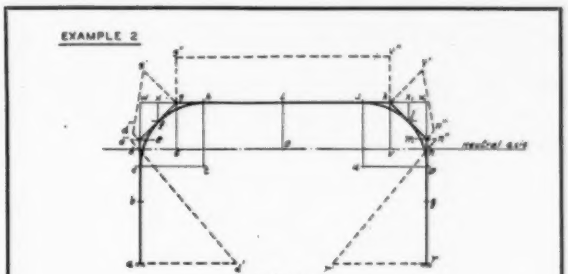


FIG. 10

Judgment must be used in determining the allowable working stresses. Fig. 9 gives an indication of the manner in which the ultimate strength, yield point and proportional limit vary with the temperature. The authors prefer to base their calculations on the proportional limit rather than on the yield point, as the former decreases with increase of temperature, while the latter increases to a maximum before its value commences to decrease. If the working stresses are based on the yield point or ultimate strength, there is a possibility of going outside the range within which Hooke's Law[4] holds; then the fundamental premise of the strength of materials is not valid.

The pipe lines of examples 2 to 7, inclusive, have their ends fixed. The examples are self-explanatory.

EXAMPLE 2, FIG. 10. Taking moments about *gk*, for the centroid

$$\begin{aligned} 80 \times 0 &= 0 \\ 2 \times (20 \times 7) &= 280 \\ 2 \times (46 \times 37) &= \underline{3404} \\ & 3684 \end{aligned} \qquad 80 + 40 + 92 = 212 \qquad \frac{3684}{212} = 17.4 \text{ in.} = io$$

For the denominator of equation (11),

$$\text{for triangles } add' \text{ and } rnn' = 2 \times \frac{(42.6)^2 \times 42.6}{3} = 51{,}540$$

$$\text{for triangles } edd'' \text{ and } mnn'' = 2 \times \frac{(3.4)^2 \times 3.4}{3} = 26$$

$$\text{for trapezoids } gs'd'''e \text{ and } kv'n'''m = 2\left[\frac{(17.4)^2 + 17.4 \times 3.4 + (3.4)^2}{3}\right]20 = 4{,}980$$

$$\text{for rectangle } gs''v''k = (17.4)^2 \times 80 = \underline{24{,}221}$$

$$80{,}767$$

$$\Delta = \frac{108}{12} \times \frac{4.012}{100} = 0.362 \text{ in.}$$

$$P = \frac{0.362 \times 26{,}900{,}000 \times 3.58}{80{,}767} = 433 \text{ lb.}$$

$$M_a = M_r = 433 \times 42 = 18{,}190 \text{ in.-lb}$$

$$S = \frac{18{,}190 \times 3.25}{2 \times 3.58} = 8250 \text{ lb per sq in.}$$

The total longitudinal stress equals 8250 + 320 = 8570 lb per sq in.

EXAMPLE 3, FIG. 11. Taking moments about c_1f_1,

$$\begin{aligned} 92 \times 0 &= 0 \\ 2 \times (20 \times 7) &= 280 \\ 58 \times 43 &= 2494 \\ 28 \times 28 &= \underline{784} \\ & 3558 \end{aligned} \qquad 92 + 2 \times 20 + 58 + 28 = 218 \qquad \frac{3558}{218} = 16.3 \text{ in.} = ol$$

Taking moments about ac_1,

$$\begin{aligned} 58 \times 0 &= 0 \\ 20 \times 7 &= 140 \\ 92 \times 60 &= 5520 \\ 20 \times 113 &= 2260 \\ 28 \times 120 &= \underline{3360} \\ & 11{,}280 \end{aligned} \qquad \frac{11{,}280}{218} = 51.7 \text{ in.} = ou$$

[4] At 950 F and above, it is customary to use pipes made of carbon molybdenum steel, data for which should be obtained from the manufacturer.

The denominator of equation (11) is

for triangle	acv'	$= \frac{1}{3} \times (42)^2 \times 43$	= 25,284
" "	ecw'	$= \frac{1}{3} \times (15)^2 \times 15$	= 1125
" "	kb_1y''	$= \frac{1}{3} \times (3)^2 \times 3$	= 9
" "	mb_1a_1'	$= \frac{1}{3} \times (14)^2 \times 17$	= 1111
for trapezoid	$egx'w''$	$= \frac{1}{3} \times \left[(15)^2 + 15 \times 24 + (24)^2\right] \times 20$	= 7740
" "	$gky'x''$	$= \frac{1}{3} \times \left[(24)^2 + 24 \times 3 + (3)^2\right] \times 92$	= 20,148
" "	$ma_1''z_1'q$	$= \frac{1}{3} \times \left[(14)^2 + 14 \times 42 + (42)^2\right] \times 28$	= 23,748
			79,165

$$\Delta = \frac{124}{12} \times \frac{4.012}{100} = 0.415 \text{ in.}$$

$$P = \frac{0.415 \times 29{,}600{,}000 \times 3.58}{79{,}165} = 556 \text{ lb}$$

$$M_a = M_q = 556 \times 42 = 23{,}352 \text{ in.-lb}$$

$$S = \frac{23{,}352 \times 3.25}{2 \times 3.58} = 10{,}600 \text{ lb per sq in.}$$

The total longitudinal stress equals 10,600 + 320 = 10,920 lb per sq in.

EXAMPLE 3

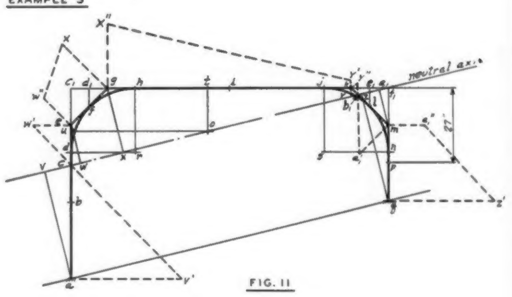


FIG. 11

ab = be = 29" ; ad = 48" ; ae = 58" , ac = 49" , kb₁ = 3" ; b₁m = 17" , ec = 15" ; ef = fg = kl = lm = 10" ; eg = km = 20" ; gk = 92" ; hj = 72" ; mp = pq = 14" ; mq = 28" ; qn = 18" ; radius rd = sn = 24" ; ot = 16.6" ; ou = 51.6" ; av = av' = 42" ; ew = ew' = ew" = 15" ; gx = gx' = gx" = 24" ; ky = ky' = ky" = 3" ; ma₁ = ma₁' = ma₁" = 14" ; qz = qz' = 42" ; aq = 124" ; c₁d₁ = fd₁ = le₁ = 7" ; bc₁ = 49" ; ic₁ = 60" ; ec₁ = 119" ; f₁c₁ = 120"

EXAMPLE 4, FIG. 12. Taking moments about ae_1,

$60 \times 0 = 0$	$60 + 17 + 116 + 24 + 32 = 249$
$17 \times 5 = 85$	
$116 \times 66 = 7656$	$\frac{15{,}421}{249} = 62 \text{ in.} = oc_1$
$24 \times 132 = 3168$	
$32 \times 141 = 4512$	
15,421	

Taking moments about ak_1,

$60 \times 30 = 1800$	
$17 \times 66 = 1122$	
$116 \times 88 = 10{,}208$	$\frac{17{,}738}{249} = 71.3 \text{ in.} = og_1$
$24 \times 96 = 2304$	
$32 \times 72 = 2304$	
17,738	

The denominator of equation (11) is

for triangle	aa_1t'	$= \frac{1}{3} \times (43)^2 \times 46$	= 28,351
" "	da_1u_1	$= \frac{1}{3} \times (13)^2 \times 14$	= 788
" "	jkw''	$= \frac{1}{3} \times (7)^2 \times 9$	= 147
" "	kmx_1	$= \frac{1}{3} \times (13)^2 \times 15$	= 845
for trapezoid	$dfv'u''$	$= \frac{1}{3} \times \left[(13)^2 + 13 \times 21 + (21)^2\right] \times 17$	= 5004
" "	$fjw'v''$	$= \frac{1}{3} \times \left[(21)^2 + 21 \times 7 + (7)^2\right] \times 116$	= 24,630
" "	$qmx''z'$	$= \frac{1}{3} \times \left[(13)^2 + 13 \times 43 + (43)^2\right] \times 32$	= 27,488
			87,253

$$\Delta = \frac{151}{12} \times \frac{4.012}{100} = 0.505 \text{ in.}$$

$$P = \frac{0.505 \times 29{,}600{,}000 \times 3.58}{87{,}253} = 614 \text{ lb}$$

$$M_a = M_q = 614 \times 43 = 26{,}400 \text{ in.-lb}$$

$$S = \frac{26{,}400 \times 3.25}{2 \times 3.58} = 11{,}900 \text{ lb per sq in.}$$

EXAMPLE 4

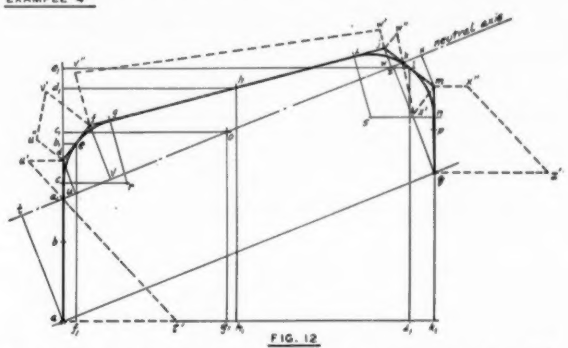


FIG. 12

ab = bd = 30" ; aa₁ = 46" ; a₁d = 16" ; ac = 58" ; ad = 60" ; de = ef = 8.5" ; df = 17" ; gi = 96" ; fh = hj = 58" ; fj = 116" ; jl = lm = 18" ; jm = 24" ; jk = 9" ; km = 15" ; mp = pq = 16" ; mq = 32" ; nq = 21" ; aq = 151" ; radius rc = sn = 24" ; et = et' = 46" ; du = du' = du" = 13" ; fv = fv' = fv" = 21" ; jw = jw' = jw" = 7" ; mx = mx' = mx" = 13" ; qz = qz' = 43" ; oc₁ = 62" ; og₁ = 71.3" ; eb₁ = 5" ; ef₁ = 66" ; hd₁ = 66" ; hh₁ = 88" ; le₁ = 132" ; li₁ = 96" ; pk₁ = 72" ; ak₁ = 141"

The total longitudinal stress is 11,900 + 320 = 12,220 lb per sq in.

EXAMPLE 5, FIG. 13-*a*. It is apparent that the centroid, o, is located at h, the point of symmetry. The denominator of equation (11) is

for triangles	adt' and plq''	$= 2 \times \frac{1}{3} \times (31)^2 \times 46$	= 29,470
" "	fhs'' and jhr'	$= 2 \times \frac{1}{3} \times (30)^2 \times 40$	= 24,000
" trapezoids	$dfs't''$ and $jlq'r''$	$= 2 \times \frac{1}{3}\left[(31)^2 + 31 \times 30 + (30)^2\right] \times 20$	= 37,210
			90,680

$$\Delta = \frac{162}{12} \times \frac{4.012}{100} = 0.542''$$

$$P = \frac{0.542 \times 29{,}600{,}000 \times 3.58}{90{,}680} = 640 \text{ lb}$$

$$M_a = M_h = M_p = 0$$

$$M_d = M_l = 640 \times 31 = 19{,}850 \text{ in.-lb}$$

$$S = \frac{19{,}850 \times 3.25}{2 \times 3.58} = 9000 \text{ lb per sq in.}$$

EXAMPLE 5

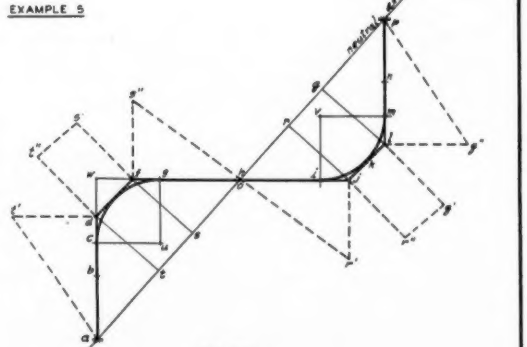


FIG. 13-a

ab = bd = ln = np = 23.5" ; ac = pm = 36" ; ad = pl = 46" ; de = ef = jk = kl = 9.5" , df = jl = 20" ; gi = 60" ; fh = hj = 40" ; fj = 80" ; radius uc = vi = 24" ; ap = 162" ; aw = 60" ; dt = dt' = dt" = lq = lq' = lq" = 31" ; fs = fs' = fs" = jr = jr' = jr" = 30"

The total longitudinal stress is 9000 + 320 = 9320 lb per sq in.

EXAMPLE 6, FIG. 13-*b*. Taking moments about ah_1,

$70 \times 0 = 0$	$70 + 20 + 92 + 20 + 28 = 230$
$20 \times 7 = 140$	
$92 \times 60 = 5520$	$\frac{11{,}280}{230} = 49 \text{ in.} = oe_1$
$20 \times 113 = 2260$	
$28 \times 120 = 3360$	
11,280	

Taking moments about ad_1,

$70 \times 35 =$	2450
$20 \times 77 =$	1540
$92 \times 84 =$	7728
$20 \times 91 =$	1820
$28 \times 112 =$	3136
	16,674

$$\frac{16{,}674}{230} = 72.4 = ob_1$$

EXAMPLE 6

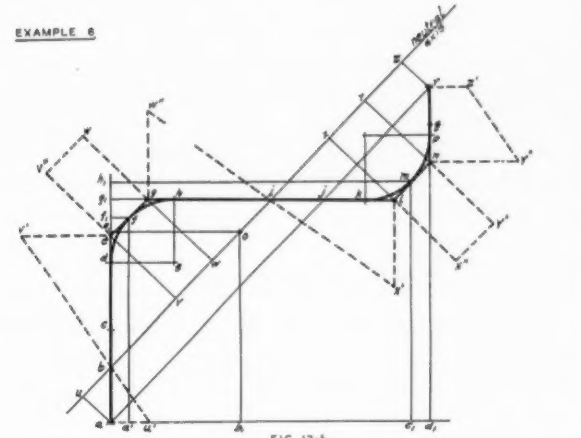


FIG. 13-b

$ac = ce = 35''$; $ae = 70''$; $ad = 60''$; $ab = 21''$; $be = 49''$;
$ef = fg = lm = mn = 10''$; $eg = ln = 20''$; $hk = 72''$; $gi = il = 46''$; $gl = 92''$;
$pr = 18''$; $ng = gr = 14''$; $nr = 28''$; radius $sd = tp = 26''$; $ob_1 = 72.4''$;
$oe_1 = 49''$; $ff_1 = 7''$; $fa_1 = 77''$; $ag_1 = 84''$; $ig_1 = 60''$; $mh_1 = 113''$;
$mc_1 = 91''$; $ad_1 = 120''$; $gd_1 = 112''$; $au = au' = 14''$; $ev = ev' = ev'' = 34''$;
$gw = gw' = gw'' = 34''$; $lx = lx' = lx'' = 34''$; $ny = ny' = ny'' = 34''$;
$rz = rz' = 14''$; $ar = 174''$.

The neutral axis az cuts the line gl at the midpoint i. By using elementary analytic geometry, it can be proved that this will be so for all figures of this same general shape.

The denominator of equation (11) is

for triangle abu'	$= \frac{1}{3} \times (14)^2 \times 21$	$= 1{,}372$
" " ebv'	$= \frac{1}{3} \times (34)^2 \times 49$	$= 18{,}881$
for triangles giw'' and lix'	$= 2 \times \frac{1}{3} \times (34)^2 \times 46$	$= 35{,}450$
for rectangles $egw'v''$ and $lny'x''$	$= 2 \times (34)^2 \times 20$	$= 46{,}240$
for trapezoid $nrz''y''$	$= \frac{1}{3} \times \left[(34)^2 + 34 \times 14 + (14)^2\right] \times 28$	$= 17{,}061$
		119,004

EXAMPLE 7

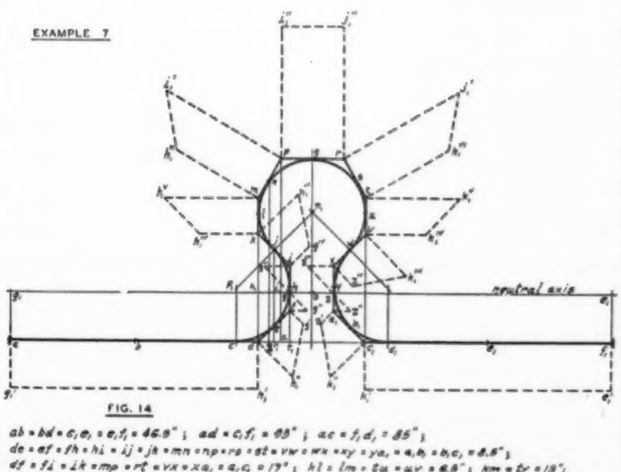


FIG. 14

$ab = bd = c_1e_1 = e_1f_1 = 46.5''$; $ad = c_1f_1 = 93''$; $ac = f_1d_1 = 85''$;
$de = ef = fh = hi = ij = jk = mn = np = rs = st = vw = wx = xy = ya_1 = a_1b_1 = b_1c_1 = 8.5''$;
$df = fi = ik = mp = rt = vx = xa_1 = a_1c_1 = 17''$; $hl = lm = tu = uv = 6.5''$; $km = tv = 13''$;
$pq = qr = 12''$; $pr = 24''$; $er_1 = 6''$; $ht_1 = 20''$; $j_1r = 35''$; $ld = 48$; $nq_1 = 62''$; $pi_1s_1 = 60''$;
$ef_1 = 227''$; radius $p_1j = jn_1 = 20''$; $ag_1 = ag_1' = c_1k_1' = e_1'f_1 = dh_1'' = c_1k_1'' = 18.8''$;
$fg = fg' = fg'' = a_1z = az' = a_1z'' = 6.5''$; $ig = ig''' = ig^{iv} = xz = xz''' = xz^{iv} = 10.5''$;
$kh_1 = kh_1''' = kh_1^{iv} = vk_1 = vk_1''' = vk_1^{iv} = 22''$; $mh_1 = mh_1^{v} = mh_1^{vi} = tk_1 = tk_1^{v} = tk_1^{vi} = 36''$;
$pi_1 = pi_1' = pi_1'' = rj_1 = rj_1' = rj_1'' = 50.5''$.

$$\Delta = \frac{174}{12} \times \frac{4.012}{100} = 0.580 \text{ in.}$$

$$P = \frac{0.580 \times 29{,}600{,}000 \times 3.58}{119{,}004} = 518 \text{ lb}$$

$$M_e = M_g = M_l = M_n = 518 \times 14 = 17{,}610 \text{ in.-lb}$$

$$S = \frac{17{,}610 \times 3.25}{2 \times 3.58} = 8000 \text{ lb per sq in.}$$

The total longitudinal stress is $8000 + 320 = 8320$ lb per sq in.

EXAMPLE 7, FIG. 14. Taking moments about af_1,

$2 \times 93 \times 0 =$	0
$2 \times 17 \times 6 =$	204
$2 \times 17 \times 20 =$	680
$2 \times 17 \times 35 =$	1190
$2 \times 13 \times 48 =$	1248
$2 \times 17 \times 62 =$	2108
$2 \times 12 \times 69 =$	1656
	7086

$$2 \times (93 + 17 + 17 + 17 + 13 + 17 + 12) = 376$$

$$\frac{7086}{376} = 18.8 \text{ in.} = ag_1 = f_1e_1$$

The denominator of equation (11) is

for triangles fgg'' and a_1zz''	$= 2 \times \frac{1}{3} \times (6.5)^2 \times 6.5$	$= 183$
" " gig''' and zxz'''	$= 2 \times \frac{1}{3} \times (10.5)^2 \times 10.5$	$= 770$
" rectangles $ag_1'h_1'd$ and $c_1k_1'e_1'f_1$	$= 2 \times (18.8)^2 \times 93$	$= 65{,}739$
" rectangle $prj_1''i_1''$	$= (50.5)^2 \times 24$	$= 61{,}206$
for trapezoids $kig^{iv}h_1'''$ and $xvk_1'''z^{iv}$	$= 2 \times \frac{1}{3}\left[(10.5)^2 + 10.5 \times 22 + (22)^2\right] \times 17$	$= 9{,}353$
" " $kmh_1^{v}h_1^{iv}$ and $tk_1^{v}k_1^{iv}v$	$= 2 \times \frac{1}{3}\left[(22)^2 + 22 \times 36 + (36)^2\right] \times 13$	$= 22{,}290$
" $mpi_1'h_1^{vi}$ and $rj_1'k_1^{vi}t$	$= 2 \times \frac{1}{3}\left[(50.5)^2 + 36 \times 50.5 + (36)^2\right] \times 17$	$= 64{,}195$
		223,736

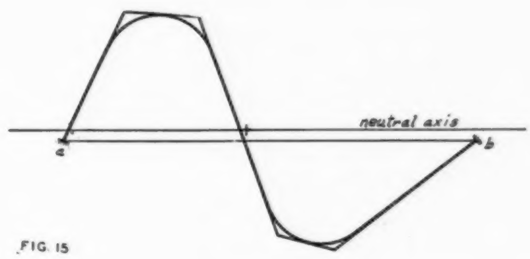


FIG. 15

$$\Delta = \frac{227}{12} \times \frac{4.012}{100} = 0.76 \text{ in.}$$

$$P = \frac{0.76 \times 29{,}600{,}000 \times 3.58}{223{,}736} = 360 \text{ lb}$$

$$M_q = 360 \times 50.5 = 18{,}200 \text{ in.-lb}$$

$$S_q = \frac{18{,}200 \times 3.25}{2 \times 3.58} = 8250 \text{ lb per sq in.}$$

The total longitudinal stress is $8250 + 320 = 8570$ lb per sq in.

The method of procedure given above can also be applied to pipes which have more complicated shapes than those already given. An example is the shape shown in Fig. (15).

COMBUSTION ENGINEERING COMPANY, INC., has opened its own branch office in Birmingham, Ala., under the direction of J. B. Emory as District Manager, assisted by W. E. Johnson. Mr. Emory has been associated with Combustion Engineering Company for several years in shop management and sales capacities, and Mr. Johnson has long been identified with the sales of this company's equipment in the Birmingham territory. The address is 516 Martin Building, Birmingham, Ala.

REVIEW OF NEW BOOKS

Any of the books reviewed on this page may be secured from Combustion Publishing Company, Inc., 200 Madison Ave., New York

Combustion, Flames, and Explosion of Gases

By Bernard Lewis and Guenther von Elbe

The purpose of this book is perhaps best expressed by the following quotation from the preface: "The interest in combustion phenomena, shared alike by chemists, physicists and engineers, has led to a large number of investigations which vary widely in purpose and accomplishment as well as in actual scientific value. There has been need of coordination and critical appraisal of these recorded investigations. This the authors have attempted to do."

The text is divided into four parts, namely, (1) chemistry and kinetics of the reactions between fuel gases and oxygen; (2) propagation of flames, in which are discussed the influence of vessel shape and gas motion, flame photography and theory of the burning velocity; (3) state of the burnt gas; and (4) problems in technical combustion processes.

Those concerned with design involving basic combustion phenomena will find the book helpful.

The book contains 415 pages, is cloth bound and priced at $5.50.

Modern Furnace Technology

By H. Etherington

While the major portion of this book deals with the various phases of industrial furnace design, rather than furnaces of steam generating units, much of the text is devoted to fundamental considerations such as the theory of gas flow, combustion of gases, liquid fuels and coal, the principles of heat transfer, heat exchangers, refractory materials and physico-chemical considerations. Many problems are worked out to illustrate application of the formulas given. The text is amply illustrated and useful charts are appended.

There are 524 pages, 6 x 9 in., with cloth binding; price $12.

Air Conditioning, Heating and Ventilating

By J. R. Dalzell and C. L. Hubbard

This is a practical treatise of 571 pages dealing with the fundamentals of heating, ventilating and air conditioning and their application in a non-technical or "how-to-do-it" manner. Numerous typical examples are given and their solutions worked out, only arithmetic or simple algebra being employed in the calculations. Many manufacturers' tables of operation and selection have been included.

The contents include: Fundamentals of Air Conditioning, Comfort Standards, Insulation, Heating Boilers, Ventilating Systems, Radiators, Direct Steam Heating, Hot-Water Heating (both natural and forced circulation), Vacuum Systems, Fans, Firing Equipment, Automatic Controls, Air Conditioning Appliances, Cooling Methods, Air-Conditioning Units and Applications.

The price is $4.

Heat Exchange Standards

The Heat Exchange Institute announces the publication of a new section of its Standards entitled, "Steam Jet Ejector and Vacuum Cooling Section."

Part 1 dealing with steam jet ejectors covers nomenclature, operating principles, types of assemblies, capacity and standard accessories and materials of construction. Part 2, dealing with a test code for steam jet ejectors, covers motive steam, condensing water, vacuum and pressure measurement, capacity measurement, performance tests, diagrammatic arrangements of apparatus for conducting various tests, standard air nozzle orifices and curves on steam flow and air-water-vapor mixture data. Part 3 takes up steam jet vacuum refrigeration equipment and covers nomenclature, definitions, performance, construction, standard units and special types. The price is $1.

Standards on Coal and Coke

After an interval of several years there has just been issued a new revised compilation of all A.S.T.M. standardized methods of testing, specifications and definitions pertaining to coal and coke.

Standard test procedures are given covering such subjects as sampling, fineness, grindability, drop shatter test, tumbler test, screen analysis, size designations and cubic foot weight.

The specifications cover gas and coking coals, foundry coke, classification of coals by rank and by grade, and round-hole screens for testing.

The price of this 150-page publication with heavy paper binding is $1.25.

Manual of Ordinances and Requirements

This 160-page paper-bound book is compiled by the Smoke Prevention Association in the interest of those having to deal with air pollution, smoke elimination and combustion of fuel. It contains a tabulation of excerpts from the smoke ordinances of the various cities, the methods employed by them to determine smoke densities, penalties invoked and comments from smoke enforcement bodies on the problems encountered. In addition, there are descriptions of methods of measuring air pollution and a reprint of papers delivered at the last annual convention of the Association. Price, 50 cents.

FABRICATING ALLOY STEELS

By A. R. McLAIN*

Among the materials discussed are plain carbon steel, carbon-molybdenum steel, 2 to 3 per cent nickel steel, low- and high-chrome steel and 18–8 chrome-nickel steel. The influence of each of these alloys on the physical properties of the steel is reviewed, as well as the problems presented in their fabrication and welding. The applications and limitations of each are discussed.

HEAT-RESISTING and corrosion-resisting steels may be considered basically as iron-carbon alloys to which are added either chromium or chromium and nickel. These chromium steels were first used about the middle of the nineteenth century, but only in the form of mining tools and structural steel until early in the twentieth century.

In 1913, Brearley developed stainless steel cutlery with 11 to 13 per cent chrome and 0.30 to 0.40 per cent carbon. Since then chromium irons with lower carbon content have been used extensively in engineering applications. The low-carbon 18–8 chrome-nickel alloy was developed about 1912. More recent developments have been made on 18–8 chrome-nickel by using stabilizers, the most important of which are titanium and columbium.

That used most extensively in the fabrication of pressure vessels is carbon steel. This is relatively inexpensive and is readily adapted to the forming and welding operations. When design requires plates $4^1/_2$ in. thick or over, it is necessary to use material with 0.40 to 0.60 per cent molybdenum to be sure of a minimum tensile strength of 70,000 lb per sq in. Small quantities of molybdenum increase the tensile strength at 930 to 1110 F and also give the steel much improved creep strength at moderate operating temperatures.

It is more difficult to fabricate drums from carbon-molybdenum steel than from ordinary carbon steel. Pressing is more difficult as the yield strength at atmospheric temperature is approximately 30 per cent higher, and the welding is more difficult since cracking may be frequent unless a comparatively high preheat is used. Tests indicate that a preheat of 400 or 500 F is necessary to reduce the hardness and eliminate cracking.

We have also successfully welded manganese-molybdenum steel 4 in. thick having a tensile strength of 100,000 lb per sq in. and believe it feasible to weld boiler drums made from this material when it is accepted by the Code. Cromansil steel has also been successfully welded.

* Hedges-Walsh-Weidner Division, Combustion Engineering Company, Chattanooga, Tenn.

For operating temperatures above 900 F or 1100 F, carbon steel is not suitable, due to its high scaling rate. Dean MacQuigg recently gave some data on the rate of oxidization of plain carbon steel containing 0.15 to 0.20 per cent carbon, when exposed for one week to an oxidizing atmosphere at increasingly higher temperatures. The specimens used were cubes approximately 0.5 in. on a side. After a given exposure, the adherent scale was flaked off as completely as possible without removing appreciable metal. The cubes were weighed after each period of heating, and the loss was calculated and expressed in percentage of the original weight. Results were as follows:

Temperature, F	Loss in Weight One Week—Per Cent
570	0.09
750	0.22
890	0.52
1020	0.43
1200	3.33
1380	15.10
1560	36.80
1830	100.00

From the foregoing tabulation, one can readily see that some material other than carbon steel must be used for high-temperature operation. Before choosing the material, it is necessary to know the nature of the process, operating conditions, cost, ease of fabrication and any other facts that might enter into the particular application.

2–3 Per Cent Nickel

The 2–3 per cent nickel is extensively used for low-temperature operation. This material has good physical properties at sub-zero temperatures, and is fairly easy to fabricate. We have been welding this material for five or six years and have had very little operating difficulty. It is acceptable by the A.S.M.E. Code for sub-zero temperature operation.

4–6 Per Cent Chrome

This material is used extensively in the oil industry in the form of tubular products, and also for superheater tubes. It shows a resistance to sulphide corrosion four to ten times that of ordinary steel, and resistance to oxidation at 1000 F three times as great.

In fabricating vessels from plate, the greatest obstacle to overcome is its intense air-hardening properties. A preheat of about 500 F is necessary before welding is started, and the finished welded vessel must be annealed without allowing it to cool in order to prevent cracking of the weld and affected area. This material should not be used in service for temperatures ranging from 900 to 1400 F except in the fully annealed state, since long-time service at these temperatures gives the same effect as annealing, and the design must be based on the annealed strength.

The addition of 0.5 per cent molybdenum or 1 per cent

tungsten increases the strength of this material at elevated temperatures considerably, and increases resistance to certain types of corrosion. Also, the addition of certain amounts of titanium and columbium renders these steels substantially non-hardening. When titanium and columbium are added in sufficient quantities, the chromium that ordinarily combines with the carbon is left in solid solution with the iron to impart corrosion- and scale-resisting characteristics.

15–16 Per Cent Chrome

The chemical composition of this alloy is the primary factor in corrosion resistance. The mechanical condition is also important, hence processing and fabrication is vitally important from the standpoint of corrosion resistance and physical properties. Due to certain peculiarities, it requires special handling to insure satisfactory results. It should not be taken above 1550 or 1600 F during fabrication, unless mechanical work is to be done, as excessive grain growth and weakening of physical properties takes place above this temperature.

The air-hardening of this material is overcome by preheating to 300 or 400 F for welding. After the outside of the weld is completed, it is necessary to anneal the vessel to soften the weld so that the back chip can be made without cracking the weld. Completed vessels are annealed by holding at 1400 to 1450 F for four hours per inch of thickness, but not less than four hours, and cooling in the furnace at a rate not exceeding 200 deg per hour to 1000 F. After annealing, all welds are X-rayed and the entire vessel is sandblasted inside and out and pickled with 20 per cent nitric acid at 120 F.

18–8 Per Cent Chrome-Nickel

The range of austenitic steel ordinarily used in commercial production in America is carbon under 0.20 per cent, manganese under 0.50 per cent, chromium 17–20 per cent and nickel 7 to 10 per cent. These steels cannot be hardened by heat treatment, and possess their maximum softness and corrosion resistance after annealing at 1800 to 2000 F and cooled to 800 F in less than 4 minutes. They are completely stable up to approximately 800 F, but between 800 F and 1600 F, when the carbon is above 0.02 per cent, carbide precipitation occurs along the grain boundaries.

These materials are easily fabricated, but are limited to Class 1 vessels where heavy sections are required, due to inability to give the finished vessel proper heat treatment. To prevent these chrome carbides from forming along the grain boundaries, titanium (6 times the carbon) and columbium (10 times the carbon) have been added as stabilizers. These stabilizing elements unite with the carbon and leave the chromium in suspension. Of these two stabilizers, columbium is the more practical from a welding standpoint, as it can be deposited in weld metal.

Comparatively thick plate can be fabricated into vessels by using stabilized materials. This enables the vessels to be heat-treated at 1800 to 1950 F, and air cooled to 800 F in 20 to 30 min, which is rapid enough to prevent carbide precipitation.

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Heat Transmission in Combustion Chambers

The November 1938 issue of *The Steam Engineer*, London, contains an article by W. C. Carter entitled, "Heat Transmission in Combustion Chambers and its Effect on Design with Bare Tube Water Walls." This was the winning paper in a staff competition of Messrs. John Thompson Ltd., well-known builders of water-tube boilers in England.

The formula employed to express heat transfer by radiation to the furnace walls is based on a modification of the Stefan-Boltzmann equation and is as follows:

$$R = 0.172 \times A \left\{\left(\frac{T_F}{100}\right)^4 - \left(\frac{T_{TW}}{100}\right)^4\right\} \alpha \times \delta$$

where R is the net rate of heat radiation in Btu per hr; A is the projected area of heating surface; T_F the absolute furnace temperature in degrees F; T_{TW} is the absolute wall temperature of the tubes forming the heating surface; α is the correction to allow for the absorptivity of the heating surface and the emissivity of the fuel undergoing combustion; and δ is a correction for the disposition of the heating surface in relation to the manner in which it "sees" the furnace.

For different methods of firing the same basic formula is employed, the only modification being in the factor δ.

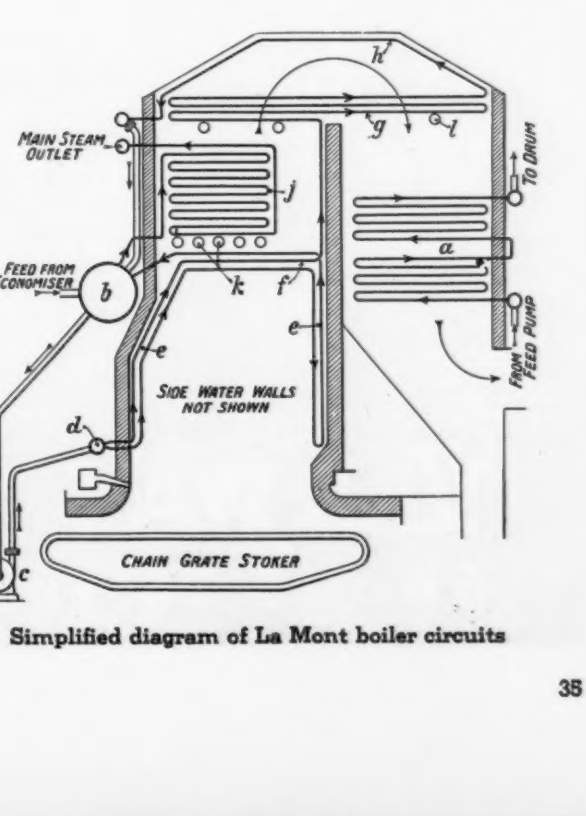
After pointing out that with refractories, once more or less stabilized conditions have become established, that the heat added by radiation and convection is practically equal to the loss from the refractory, the author makes the following comments:

"It is now becoming more appreciated that furnace temperature is the predominating influence in combustion chamber design and that heat release rate is purely and simply a guide or starting point, this fact is illustrated by the high heat release rates used with success in gas- and oil-fired boilers, and in stoker- and pulverized fuel-fired boilers it has been found by this firm that, provided the combustion chamber temperature is kept within reasonable limits below the fusion temperature of the ash, no great difficulty with regard to refractory maintenance, slagging or ignition will be experienced."

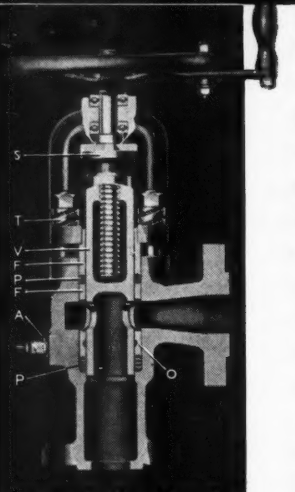
"Sometimes the danger of overcooling the combustion chamber is advanced by advocates of the block-covered type of wall; but in the light of recent experience and research, it has been established beyond question that if bare tube water-cooled furnaces have any drawbacks, delayed combustion is not one of them. The whole question of ignition boils down to a parity with the starting of a cold motor-car engine where the air supply is choked to obtain a rich, readily ignitible mixture. This remark similarly applies to the starting up of a cold boiler or the maintenance of ignition at any load on the boiler. Only sufficient air for ignition of the fuel should be supplied in contact with the fuel during the actual ignition period; the additional air for complete combustion should be supplied outside the zone of ignition. Beyond the temperature required for quick ignition of the fuel it has been found that nothing is gained by increasing the furnace temperature, in fact rather the reverse holds. With coal it has been found experimentally that satisfactory ignition takes place at about 700 C (1292 F) and that the rate of combustion is diminished, not increased with rising furnace temperature. Also, for coal, it has been found that increasing the furnace temperature from 700 C to 1000 C (1832 F) decreased the rate of combustion by approximately 40 per cent, which corrects the impression held by a number of engineers that reduction in temperature retards combustion, and strongly supports the advantages already advocated by this firm for bare-tube type water walls."

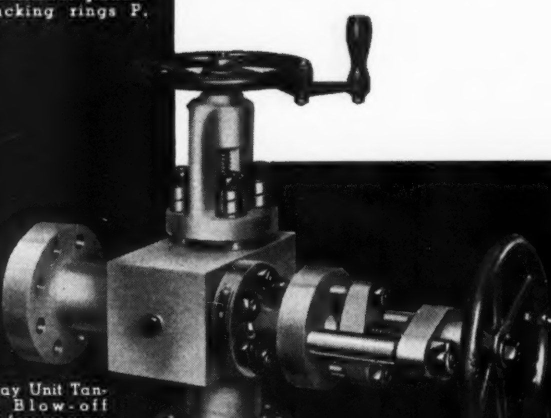
British Industrial Installs La Mont Topping Unit

The firm of G. & J. Weir, Ltd., of Glasgow, makers of pumps and other auxiliaries, has recently topped its existing 350-lb plant by a 50,000-lb per hr, 1000-lb pressure La Mont boiler and a high-pressure turbine taking steam at 850 lb, 850 F, at the throttle and exhausting to the low-pressure system. In addition to thus supplying additional power to the works, the new boiler will furnish steam for routine testing purposes and for development work in the high-pressure field. The special reasons for selecting this type of boiler were its compactness and a high degree of flexibility in operation; also, the fact that the circulation is subdivided and controlled so that each



Simplified diagram of La Mont boiler circuits





tube receives the correct amount of water in proportion to its heat-absorbing capacity was believed to contribute to its reliability.

The accompanying simplified diagram of the unit is taken from a descriptive article in the October 1938 issue of *The Power and Works Engineer*, London. The unit comprises an economizer *a* in two sections, from which the feed is led to the steam drum *b*. From this drum the water flows to the circulation pump *c*, by which it is forced to the distributing header *d*. From this header it is distributed by means of the metering nozzles to the steam-generating tubes forming the first evaporative zone which includes the radiant section *e* (the water walls surrounding the combustion chamber on four sides) and the convection section *f*, comprising several rows of tubes immediately above the combustion chamber. The mixture of steam and water formed in one portion of the evaporating tubes is led back to the drum *b*, and the remaining tubes extend up to the center division wall to form a second evaporative zone *g* and the roof tubes *h*. From the latter the mixture discharges into a collector header, thence into the steam drum. The superheater, of orthodox MeLeSco design, is located above the first evaporative zone.

Fusion welding has been employed throughout the fabrication of the boiler, including the steam drum. The unit is fired by a chain-grate stoker.

The heating surface of the boiler and water walls is given as 3050 sq ft, that of the economizer as 5380 sq ft and that of the superheater as 1750 sq ft.

On test the unit showed an overall thermal efficiency of 83.96 per cent, or 82.92 per cent net, after making deductions for the auxiliaries. Although it has been in service only a few months it has demonstrated its flexibility in handling load swings from 8000 to 38,000 lb per hr in one minute without difficulty, the water level remaining stable within 1 in. and the steam pressure and temperature being only slightly affected.

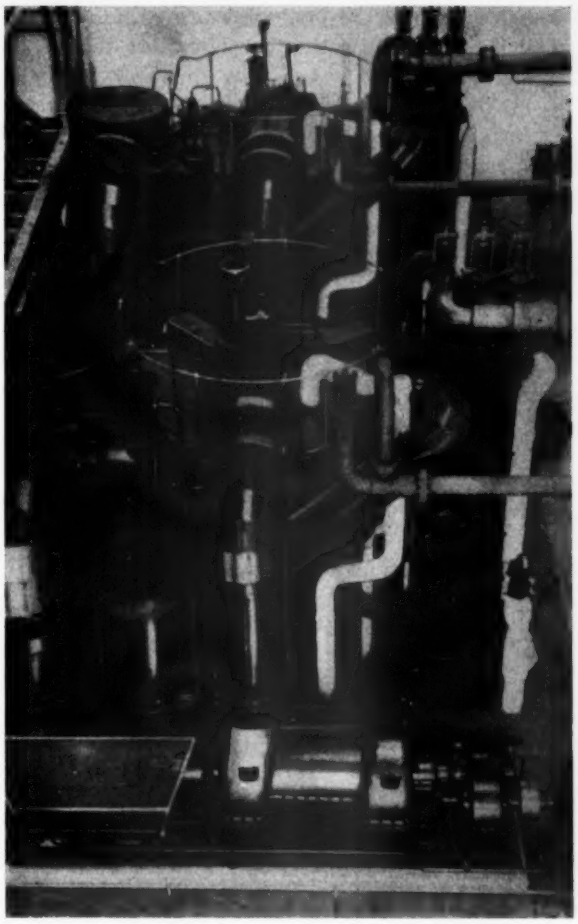
Velox Installation Completed in New Zealand

The accompanying illustration, reproduced from the November 1938 issue of *Engineering and Boiler House Review*, London, shows the two Velox steam generators recently installed at the Evans Bay Station, Wellington, New Zealand. As previously noted in these columns, each of these units has a normal rating of 90,000 lb of steam per hour at 225 lb pressure and 600 F total steam temperature. They supply steam to a 15,000-kw turbine-generator and have been designed to pick up load rapidly in view of the standby character of the station. In fact, according to the current article, they have been brought up to load from a cold condition in about four minutes.

A feature of the Velox boiler, it will be recalled, is the high rate of heat transmission due to the extremely high velocity of both the products of combustion under pressure and the water within the tubes. The combustion chamber is cylindrical with the gases passing over tubular evaporative elements arranged around the wall. Oil firing is employed and combustion takes place under pressure up to 35 lb per sq in. abs, air for combustion being supplied from a blower which is driven by a gas

turbine operated by the velocity of the gases leaving the boiler.

At full load the temperature of the products of combustion entering the gas turbine is given as 920 F for the New Zealand installation. A reduction of approximately 250 deg and 16 lb pressure takes place within the turbine, so that the gases enter the enconomizer at about 670 F.



Two 90,000-lb per hr units in Evans Bay Station

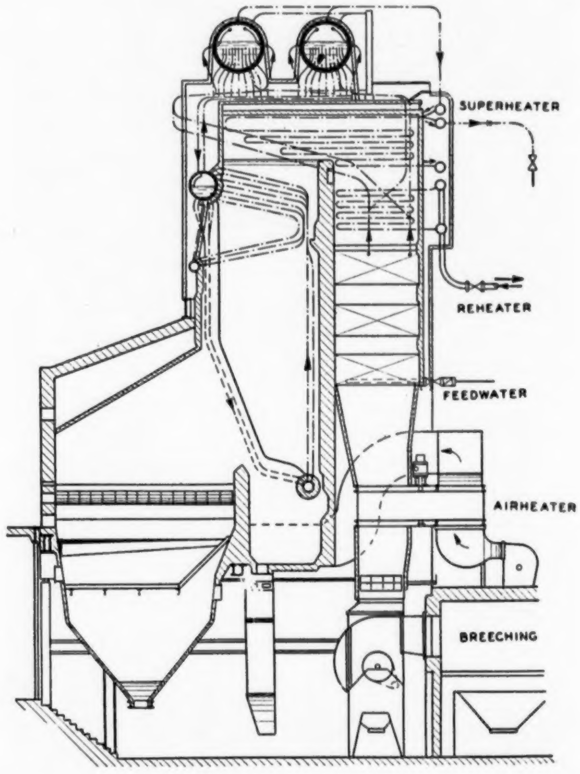
Each boiler contains only 9000 lb of water at the normal working level, or one-tenth the rated hourly output. This water is circulated at the rate of 15,000 lb per min, or ten times the maximum evaporation rate. The resultant water velocity is 20 ft per sec and that of the products of combustion on the gas side of the tubes is 600 ft per sec.

Tests on a Schmidt High-Pressure Boiler

Dr. Ing. P. Engel in *Die Wärme* of September 3, 1938, reports the results of tests conducted on one of ten Schmidt-Hartmann high-pressure boilers in a large power plant in central Germany. This type of steam generator, as will be seen from the sketch, employs a primary boiler containing pure water the steam from which gives up its latent heat and some of its sensible heat through coils to the water in the steam drum. This evaporates the water in the drum to produce steam in the secondary circuit for external use. The water in the primary circuit is condensed and recirculated.

With this arrangement there can be no scale formation within the primary circuit and the only deposit possible is on the outside of the coils within the steam drum. However, since these do not exceed the temperature of the primary steam they are not endangered through overheating.

The primary circuit was designed for a maximum pressure of approximately 2500 lb per sq in. in order to provide for sufficient heat transfer in the event that the coils in the steam drum might become coated with scale, but this precaution has thus far not been utilized in operation. Actually, the pressure in the primary circuit is kept under 2000 lb per sq in. and that of the secondary steam is slightly under 1500 lb. The heating surface of the coils within the steam drum is 1830 sq ft.



Sketch of Schmidt-Hartmann boiler showing primary and secondary circuits

TEST PERFORMANCE

	Fractional Load	Normal Load		Peak Load
1. Temperature of water leaving economizer, F	565	577	586	576
PRIMARY STEAM:				
2. Saturated steam pressure, lb per sq in.	1670	1910	1970	1990
3. Saturated steam temperature, F	610	630	633	635
4. Cooling of condensate below saturation, deg F	12.6	9	10.8	9
5. Heat given up per pound of primary steam, Btu	542	493	484	478
6. Quantity of primary steam, lb per hr	56,300	86,200	88,100	95,600
7. Ratio of primary to secondary steam, per cent	113	120.8	119	122
8. Primary water circulated per hour (number of times)	9.2	14.7	15.2	16.6
9. Loading of primary heating surface				
(a) Evaporation, lb per sq ft per hr	22.13	33.9	34.7	37.7
(b) Btu per sq ft per hr	12,000	16,700	16,800	17,950
10. Evaporation per lb of coal, lb	3.5	3.34	3.41	3.43
OPERATING STEAM				
11. Generated in drum, lb per hr	49,800	71,400	73,900	78,500
12. Steam pressure	1465	1465	1465	1465
13. Saturated steam temperature, F	584	585	588	587
14. Superheated steam temperature, before water injection, F	1003	1047	1027	1074
15. Superheated steam temperature after water injection	900	900	890	892
16. Efficiency of unit (gross), per cent	84.63	85.03	85.20	87.52

The tests, from which some figures are here quoted, provide a general idea of the thermal operation of the unit. It will be noted that because of the difference in the steam pressures within the primary and secondary circuits that the condensate is from 9 to 12.6 deg below the saturation temperature of the primary steam. The superheat temperature of the secondary steam was considerably higher than desired, hence had to be reduced to the operating value by water injection. The heat given up per pound of primary steam ranged from 478 to 542 Btu and the water in the primary circuit was evaporated, under the load range covered by the tests, from about nine to seventeen times per hour.

The article shows the distribution of work done by the primary boiler, economizer, superheater and reheater at various loads. These figures ranged, in the order named, from 41.4 to 46.5 per cent, from 18.5 to 20.8 per cent, from 25.6 to 27.1 per cent and from 9.4 to 10.7 per cent.

British Power Supply

An interesting slant on the power situation in England is contained in *The Electrical Times* of November 18 which comments editorially on the pending electricity supply bill, in part, as follows:

"Politics have always been the curse of the electricity supply industry. . . . The McGowan Committee was appointed in July 1935. Its report, produced a year later, fluttered the electricity dovecots by whole-hog recommendations which bid fair to affect the personal careers of thousands of station engineers. Big municipalities, small municipalities, power companies and distribution companies have all been pulling strings and the Ministry of Transport has had to water down the original proposals materially in view of Parliamentary support which the various interests could command."

The British Prime Minister has lately announced that the bill will not be presented at this session; hence the electrical industry will have a respite, pending the balance of conservative and liberal opinion that is likely to prevail in Parliament.

Continuing, the editorial observes that, "The recent war scare has introduced new factors and there is likely to be an increasing demand for partial decentralization; for it is realized that the superstation-plus-grid must be supplemented by local self-contained sources of supply on purely military grounds. Local administration of comparatively small areas is thus receiving a boost as against central control."

"In a couple of years, given cooperation and good will, the industry could set its house in order and reduce the need for the bill, as well as its scope."

From the foregoing it would appear that, despite reports to the contrary which occasionally reach this side of the Atlantic, all is not serene in the British power situation, and that the United States is not alone in the experience of having power made a football of local and national politics.

ADVERTISERS IN THIS ISSUE